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THESIS

PERFORMANCE OF A MULTIPLE NOZZLE EXHAUST GAS EDUCTOR SYSTEM FOR GAS TURBINE POWERED SHIPS.

by

Richard Stewart Shaw

December 1980

Thesis Advisor: Paul F. Pucci

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Performance of a Multiple Nozzle Exhaust Gas Eductor System for Gas Turbine Powered Ships

by

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Submitted in partial fulfillment of the requirements for the degree of

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ABSTRACT

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NOMENCLATURE

English Letter Symbols

A	-	Area, in.^{2}
c	-	Sonic velocity, ft/sec
C	-	Coefficient of discharge
D	-	Diameter, in.
Fa	-	Thermal expansion factor
F_{fr}	-	Wall skin-friction force, lbf
g_c	-	Proportionality factor in Newton's Second Law, g_c = 32.174 lbm-ft/lbf-sec^2
h	-	Enthalpy, Btu/bm
k	-	Ratio of specific heats
L	-	Length, in.
P	-	Pressure, in. H_2O : Plug Length
P_a	-	Atmospheric pressure, in. Hg
P_v	-	Velocity head, in. H_2O
PMS	-	Static pressure along length of mixing stack in. H_2O
R	-	Gas constant for air, 53.34 ft-lbf/lbm - R
s	-	Entropy, Btu/lbm - R
S	-	Primary dimension of mixing stack
T	-	Absolute temperature, R
u	-	Internal energy, Btu/lbm
U	-	Velocity, ft/sec
v	-	Specific volume, ft^3/lbm

W - Mass flow rate, lbm/sec

D - Distance from primary nozzle exit to mixing stack, in.

Y - Expansion factor

<u>Dimensionless Groupings</u>

A* - Secondary flow area to primary flow area ratio

AR - Area ratio

f - Friction factor

K - Flow coefficient

K_e - Kinetic energy correction factor

K_m - Momentum correction factor at the mixing stack exit

K_p - Momentum correction factor at the primary nozzle exit

M - Mach number

ΔP^* - Pressure coefficient

PMS* - Mixing stack pressure coefficient

Re - Reynolds number

D/S - Standoff; Ratio of distance from primary nozzles to entrance of mixing stack, (D) to primary dimension of mixing stack (S)

T* - Absolute temperature ratio of the secondary flow to primary flow

T_t^*=TT - Absolute temperature ratio of the tertiary flow to primary flow

W*=W* - Secondary mass flow rate to primary mass flow rate ratio

W*=WT* - Tertiary mass flow rate to primary flow rate ratio

PP* - Plug Pressure Coefficient

L/S	-	Ratio of mixing stack length to mixing stack primary dimension
$\rho*$	-	Induced flow density to primary flow density

Greek Letter Symbols

μ	-	Absolute viscosity, lbf-sec/ft^2
ρ	-	Density, lbm/ft^3

Subscripts

0	-	Section within secondary air plenum
1	-	Section at primary nozzle exit
2	-	Section at mixing stack exit
f	-	Film or wall cooling
m	-	Mixed flow or mixing stack
or	-	Orifice
p	-	Primary
s	-	Secondary
t	-	Tertiary (Cooling)
u	-	Uptake
w	-	Mixing stack inside wall

Computer Tabulated Data and Illustrative Plots

UMACH	-	Uptake Mach number
PA-PNZ	-	Pressure differential across secondary flow nozzles, in. H_2O
PA-PS	-	Static pressure at mixing stack entrance, in. H_2O
PA-PT	-	Static Pressure in tertiary air plenum, in. H_2O
PMS	-	Mixing stack static pressure, in. H_2O

POR	-	Static pressure at orifice, in. H_2O
DPOR	-	Pressure differential across orifice, in. H_2O
PU-PA	-	Static uptake pressure, in. H_2O
UM	-	Average velocity in mixing stack, ft/sec
UP	-	Primary flow velocity at primary nozzle exit, ft/sec
UU	-	Primary flow velocity in uptake, ft/sec
TOR	-	Air temperature at orifice, ^oF
TAMB	-	Ambient air temperature, ^oF
TUPT	-	Temperature of air in uptake, ^oF

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In a study of this nature, the advances which are made are realized only through the efforts of many individuals. Although it is impossible to acknowledge everyone who assisted in this research, it is hoped that those unnamed persons will nevertheless be aware of the author's appreciation for their assistance.

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I. INTRODUCTION

With gas turbines becoming a more popular means of powering naval vessels, special considerations need to be given to their particular air breathing and exhausting characteristics. With air-fuel ratios of four to five times that of conventional steam plants and the requirement for a relatively large amount of combustion air, a large quantity of hot exhaust gas is generated. Due to gas turbine design, these exhaust gases are at temperatures significantly above those of conventionally powered ships. A few of the problems caused by these high temperatures are thermal damage to electronic equipment located in the mast of these ships, hot gas corrosion of the mast and other superstructures located in the hot gas wake, and a significant infrared radiation signature created by the hot gas plume and hot external surfaces of the stack.

This thesis is an extension of research done by Ellin [1], Moss [2] and Lemke and Staehli [7] to determine better geometric designs for the exhaust plenum and mixing stack system of gas turbine powered naval ships.

Ellin initiated the work by constructing an eductor model testing facility consisting of an uptake, primary flow nozzle, mixing stack, a means to control and measure the primary air flow, and a means to measure the secondary air flow; see Figure 1. The primary air flow in the testing facility represents a gas turbine's hot exhaust gas.

The secondary air flow is ambient air induced into the entrance of the mixing stack by the primary air flow; see Figure 2. From Ellin's study of multiple nozzle flow systems consisting of several identical round nozzles, it was determined that four primary flow nozzles were preferable to either three or five, and that nozzle length has little or no effect on the eductor system's overall performance. Ellin then verified the independence of the one-dimensional model correlation parameters used on flow rate or Mach number. He determined that for Mach numbers from 50% to 145% of the design Mach number of 0.064, the correlation parameters suggested in the one-dimensional analysis did in fact provide good correlation of the data.

Moss' work followed and it initially consisted of verifying the one-dimensional analysis as did Ellin. He then tested the effect of the stand off distance (that distance between the exit plane of the primary flow nozzles and the entrance plane of the mixing stack). For the primary flow nozzles he tested, Moss determined that the optimum stand off distance for maximum eductor pumping was a distance equal to 0.5 diameters (0.5 D_m) of the mixing stack. An independent investigation of this, conducted by Harrell [3], confirmed Moss' findings. Moss then investigated the effects of a conical transition placed on the entrance to the mixing stack. He concluded that a straight mixing stack without an entrance transition provided a better system performance.

The study conducted by Lemke and Staehi [7] investigated the effects on the eductor system's overall performance of varying the geometric configuration of the mixing stack and changing the area of the primary flow nozzles. Their work showed that a decrease in the ratio of the area of the mixing stack to primary flow nozzles from 3.0 to 2.5 decreased uptake back pressure but reduced the pumping coefficient of the eductor. Lemke and Staehli then investigated the effects of a solid diffuser, a two-ring and a three-ring diffuser. The results of these tests showed a decrease in uptake back pressure and an improvement in the eductor's pumping capacity. They then performed tests on a ported mixing stack. Their work determined that significant air flow through the ports could provide film cooling on the inside of the mixing stack. To enhance the film cooling provided by the ported mixing stack, Lemke and Staehli placed a shroud around the mixing stack. This shroud did not degrade the pumping or mixing characteristics, yet it provided thermal shielding of the mixing stack. Their final configuration was a combination of the ported mixing stack with flow through shroud and diffuser. Lemke and Staehli concluded that this geometric configuration, combined with a ratio of the area of the mixing stack to the primary flow nozzles, of 3.0, provided the best system overall performance.

As a result of the increased usage of infrared sensors, the need to reduce the infrared signature of the mixing stack is felt to be a necessity. Of primary concern is the concealment of the hot primary flow nozzles from an overhead view. This current study sought to develop a new mixing stack geometry which would eliminate this overhead view. The proposed idea was to insert a plug, which could be cooled, in the mixing stack.

After consideration of numerous plug and mixing stack configurations, a rectangular mixing stack and symmetric plug was chosen. This decision was based on two factors; first, the rectangular geometry has the potential for allowing installation of multiple stacks in a smaller volume than a cylindrical geometry and secondly, the ease of manufacturing a model for testing. Using the design of Lemke and Staehli, a relation between the circular cross sectional flow area and the new rectangular cross sectional flow area was established, (see Appendix B). Additionally, the overall mixing stack length, standoff distance and ratio of the mixing stack area to primary flow nozzle area obtained by Lemke and Staehli was maintained. Figures 1 and 2 provide a schematic representation of the model testing facility. Figure 3 illustrates the location and the terminology used to define the air flows.

With the new mixing stack geometry, Figures 4 and 5, a new design for the primary flow nozzles was required.

The final primary flow nozzle design tested in this study is dimensionally shown in Figures 6 and 7 and pictured in Figure 8.

After location of low pressure points on the plug, the plug was modified incorporating the results of Lemke and Staehli. The plug was ported, to provide film cooling along the inside of the plug, and shrouded, to direct the flow to the ports and provide a means of convectively cooling the top half of the plug. This cooling air, which is the tertiary flow in the system, is induced ambient air. This is contrasted with secondary air flow, which is ambient air predominantly intended to reduce the exhaust gas temperature by mixing in the eductor. Figures 9, 10 and 11 show dimensionally the modifications to the plug.

Evaluation of the eductor system performance was measured in four areas: the amount of secondary air flow induced by the primary air flow, the degree of mixing of primary and induced air flows within the mixing stack system, the amount of uptake back pressure impressed upon the turbine exhaust by the eductor system, and the amount of wall cooling air available to reduce the exterior stack temperature of the eductor system.

The key factor which allows cold flow testing to predict the effects of a hot gas eductor system is the similarity of the momentum and energy transfer mechanisms in turbulent flows.

II. THEORY AND ANALYSIS

This investigation, being an extension of the work of Ellin [1] , Moss [2] and Lemke and Staehli [7], uses the same one-dimensional analysis of a simple eductior system. Similarity between the basic geometry tested by Ellin, Moss and Lemke and Staehli was maintained in order to correlate data. The dimensionless parameters controlling the flow phenomenon used by Ellin were also used in this investigation. Dynamic similarity was maintained by using Mach number similarity to establish the model's primary flow rate.

Although the analysis presented here is for an eductor model with only primary and secondary air flows, it should be kept in mind that many of the results presented are for systems with primary, secondary, and tertiary air flows. Systems with tertiary (film or wall cooling) air flows have been non-dimensionalized with the same base parameters as the secondary air flow and have been calculated using the same one-dimensional analysis. This allows for easy comparison of the results. Parameters pertaining to the secondary systems are subscripted with an "s", those relating to the tertiary are subscripted with a "t".

A. MODELING TECHNIQUE

Dynamic similarity between the models tested and the actual prototype was maintained by using the same primary air flow Mach number. For the primary air flow Mach number used (0.064), and based on the average flow properties within the mixing stack and the hydraulic diameter of the mixing stack, the air flow through the eductor system is turbulent ($Re > 10^5$). As a consequence, momentum exchange is predominant over shear interaction, and the kinetic and internal energy terms are more influential on the flow than are viscous forces. It can also be shown that Mach number represents the ratio of kinetic energy of a flow to its internal energy and is, therefore, a more significant parameter than the Reynolds number in describing the primary flow through the uptakes.

B. ONE-DIMENSIONAL ANALYSIS OF A SIMPLE EDUCTOR

The theoretical analysis of an eductor may be approached in two ways. One method attempts to analyze the details of the mixing process of the primary and secondary air streams as it takes place inside the mixing stack. This requires an interpretation of the mixing phenomenon which, when applied to a multiple nozzle system, becomes extremely complex. The other method, which was chosen here, analyzes the overall performance of the eductor system and is not concerned with the actual mixing process. The one-dimensional analysis is based on a single primary nozzle exhausting into a mixing stack, as shown in Figure 13. To avoid repetition with previous reports, only the main parameters and assumptions will be represented here. A complete derivation of analysis used can be found in Refs. 1 and 4. The one-dimensional flow analysis of the simple eductor system described depends on the simultaneous solution of the continuity, momentum and energy equations coupled with the equation of state, all compatible with specific boundary conditions.

The idealizations made for simplifying the analysis are as follows:

1. The flow is steady state and incompressible.
2. Adiabatic flow exists throughout the eductor with isentropic flow of the secondary stream from the plenum (at section 0) to the throat or entrance of the mixing stack (at section 1). Irreversible adiabatic mixing of the primary and secondary streams occurs in the mixing stack (between sections 1 and 2).
3. The static pressure across the flow at the entrance and exit planes of the mixing-tube (at sections 1 and 2) is uniform.
4. At the mixing-stack entrance (section 1) the primary flow velocity U_p and temperature T_p are uniform across the primary stream, and the secondary flow velocity U_s and temperature T_s are uniform across the secondary stream, but U_p does not equal U_s, and T_p does not equal T_s.

5. Incomplete mixing of the primary and secondary streams in the mixing stack is accounted for by the use of a non-dimensional momentum correction factor K_m which relates the actual momentum rate to the pseudo-rate based on the bulk-average velocity and density and by the use of a non-dimensional kinetic energy correction factor K_e which relates the actual kinetic energy rate to the pseudo-rate based on the bulk-average velocity and density.

6. Both gas flows behave as perfect gases.

7. Flow potential energy of position changes are negligible.

8. Pressure changes P_{s0} to P_{s1} and P_1 to P_a are small relative to the static pressure so that the gas density is essentially dependent upon temperature (and atmospheric pressure).

9. Wall friction in the mixing stack is accounted for with the conventional pipe friction factor term based on the bulk-average flow velocity U_m and the mixing stack wall area A_w.

The following parameters, defined here for clarity will be used in the following development.

$\frac{A_p}{A_m}$	area ratio of primary flow area to mixing stack cross sectional area
$\frac{A_w}{A_m}$	area ratio of wall friction area to mixing stack cross sectional area

K_p	momentum correction factor for primary flow
K_m	momentum correction factor for mixed flow
f	wall friction factor

Based on the continuity equation, the conservation of mass principle for steady flow yields

$$W_m = W_p + W_s + W_t \tag{1}$$

where

$$\begin{aligned} W_p &= \rho_p U_p A_p \\ W_s &= \rho_s U_s A_s \\ W_t &= \rho_t U_t A_t \\ W_m &= \rho_m U_m A_m. \end{aligned} \tag{1a}$$

All of the above velocity and density terms, with the exception of ρ_m and U_m, are defined without ambiguity by the virtue of idealizations (3) and (4) above. Combining equations (1) and (1a) above, the bulk average velocity at the exit plane of the mixing stack becomes

$$U_m = \frac{W_s + W_t + W_p}{\rho_m A_m} \tag{1b}$$

where A_m is fixed by the geometric configuration and

$$\rho_m = \frac{P_a}{RT_m} \tag{2}$$

where T_m is calculated as the bulk average temperature from the energy equation (9) below. The momentum equation stems from Newton's second and third laws of motion and is the conventional force and momentum-rate balance in fluid mechanics.

$$K_p\left(\frac{W_p U_p}{g_c}\right) + \left(\frac{W_s U_s}{g_c}\right) + \left(\frac{W_t U_t}{g_c}\right) + P_1 A_1 = K_m\left(\frac{W_m U_m}{g_c}\right) \tag{3}$$

$$+ P_2 A_2 + F_{fr}$$

Note the introduction of idealizations (3) and (5). To account for a possible non-uniform velocity profile across the primary nozzle exit, the momentum correction factor K_p is introduced here. It is defined in a manner similar to that of K_m and by idealization (4), supported by work conducted by Moss, it is set equal to unity. K_p is carried through this analysis only to illustrate its effect on the final result. The momentum correction factor for the mixing stack exit is defined by the relation

$$K_m = \frac{}{W_m \bar{U}_m} \int_0^{A_m} U_m^2 \rho_2 dA \tag{4}$$

where U_m is evaluated as the bulk-average velocity from equation (1b). The wall skin friction force F_{fr} can be related to the flow stream velocity by

$$F_{fr} = f\ A_w\left(\frac{U_m^2 \rho_m}{2g_c}\right) \tag{5}$$

using idealization (9). As a reasonably good approximation for turbulent flow, the friction factor may be calculated from the Reynolds number

$$f = 0.046(Re_m)^{-0.2}. \tag{6}$$

Applying the conservation of energy principle to the steady flow system in the mixing stack between the entrance and exit planes,

$$W_p\left(h_p + \frac{U_p^2}{2g_c}\right) + W_s\left(h_s + \frac{U_s^2}{2g_c}\right) + W_t\left(h_t + \frac{U_t^2}{2g_c}\right)$$

$$= W_m\left(h_m + K_e\frac{U_m^2}{2g_c}\right) \tag{7}$$

neglecting potential energy of position changes (idealization 7). Note the introduction of the kinetic energy correction factor K_e, which is defined by the relation

$$K_e = \frac{1}{W_m U_m^2} \int_0^{A_m} U_2^3 \rho_2 \, dA \tag{8}$$

It may be demonstrated that for the purpose of evaluating the mixed mean flow temperature T_m, the kinetic energy terms may be neglected to yield

$$h_m = \frac{W_p}{W_m} h_p + \frac{W_s}{W_m} h_s + \frac{W_t}{W_m} h_t \tag{9}$$

where $T_m = \phi(h_m)$ only, with idealization (6).

The energy equation for the isentropic flow of the secondary air from the plenum to the entrance of the mixing stack may be shown to reduce to

$$\frac{P_o - P_s}{\rho_s} = \frac{U_s^2}{2g_c} \tag{10}$$

similarly, the energy equation for the tertiary air flow reduces to

$$\frac{P_o - P_t}{\rho_t} = \frac{U_t^2}{2g_c}$$

The foregoing equations may be combined to yield the vacuum produced by the eductor action in either the secondary or tertiary air plenums. For the secondary air plenum, the vacuum produced is

$$P_a - P_{os} = \frac{1}{g_c A_m}\left(K_p \frac{W_p^2}{A_p \rho_p} + \frac{W_s^2}{A_s \rho_s}\left(1 - \frac{1}{2}\frac{A_m}{A_s}\right) - \frac{W_m^2}{A_m \rho_m}\left(K_m + \frac{f}{2}\frac{A_w}{A_m}\right)\right) \tag{11}$$

where it is understood that A_p and ρ_p apply to the primary flow at the entrance to the mixing stack, A_s and ρ_s apply to the secondary flow at this same section, and A_m and ρ_m apply to the mixed flow at the exit of the mixing stack system. P_a is atmospheric pressure, and is equal to the pressure at the exit of the mixing stack. A_w is the area of the inside of the mixing stack.

For the tertiary air plenum the vacuum produced is

$$P_a - P_{ot} = \frac{1}{g_c A_m}\left(K_p \frac{(W_p + W_s)}{(A_p \rho_p + A_s \rho_s)} + \frac{{W_t}^2}{A_t \rho_t}\left(1 - \frac{1}{2}\frac{A_m}{A_t}\right) - \frac{{W_m}^2}{A_m \rho_m}\left(K_m + \frac{f}{2}\frac{A_w}{A_m}\right)\right) \qquad (11a)$$

where the primary flow now consists of both the primary and secondary air flows.

C. NON-DIMENSIONAL FORM OF THE SIMPLE EDUCTOR EQUATION

In order to provide the criteria of similarity of flows with geometric similarity, the non-dimensional parameters which govern the flow must be determined. The means chosen for determining these parameters is to normalize equations (11) and (11a) with the following dimensionless groupings.

$$\Delta P^* = \frac{\dfrac{P_a - P_{os}}{\rho_s}}{\dfrac{{U_p}^2}{2g_c}}$$

a pressure coefficient which compares the pumped head $P_a - P_{os}$ for the secondary flow to the driving head $\frac{U_p^2}{2g_c}$ of the primary flow

$$\Delta PT^* = \frac{\dfrac{P_a - P_{ot}}{\rho_t}}{\dfrac{{U_p}^2}{2g_c}}$$

a pressure coefficient which compares the pumped head $P_a - P_{ot}$ for the tertiary flow to the driving head $\frac{U_p^2}{2g_c}$ of the primary flow

$W^* \quad = \quad \frac{W_s}{W_p}$ a flow rate ratio, secondary to primary mass flow rate

$WT^* \quad = \quad \frac{W_t}{W_p}$ a flow rate ratio, tertiary to primary mass flow rate

$T^* \quad = \quad \frac{T_s}{T_p}$ an absolute temperature ratio, secondary to primary

$TT^* \quad = \quad \frac{T_t}{T_p}$ an absolute temperature ratio, tertiary to primary

$\rho_s^* \quad = \quad \frac{\rho_s}{\rho_p}$ a flow density ratio of the secondary to primary flows. (note that since the fluids are considered perfect gases,

$$\rho_s^* = \frac{T_p}{T_s} = \frac{1}{T_s^*})$$

$\rho_t^* \quad = \quad \frac{\rho_t}{\rho_p}$ a flow density ratio of the tertiary or film cooling flow to primary flows. (Note that since the fluids are considered perfect gases,

$$\rho_t^* = \frac{T_p}{T_t} = \frac{1}{T_t^*})$$

$A_s^* \quad = \quad \frac{A_s}{A_p}$ an area ratio of secondary flow area to primary flow area

$A_t^* \quad = \quad \frac{A_t}{A_p}$ an area ratio of tertiary flow area to primary flow area

With these non-dimensional groupings, equations (11) and (11a) can be rewritten in dimensionless form. Since both equations follow the same format, only the results for the secondary air plenum will be presented here.

$$\frac{\Delta P^*}{T^*} = 2\,\frac{A_p}{A_m}\Big\{\Big[K_p - \frac{A_p}{A_m}\beta\Big] - \Big[W^* K_p + T^*\Big]\frac{A_p}{A_m}\beta$$

$$+ W^{*2}T^*\Big[\frac{1}{A^*}\Big(K_p - \frac{A_m}{2A^*A_p}\Big) - \frac{A_p}{A_m}\,\beta\Big]\Big\} \qquad (12)$$

where

$$\beta = K_m + \frac{f}{2}\,\frac{A_w}{A_m}\,.$$

This may be rewritten as

$$\frac{\Delta P^*}{T^*} = C_1 + C_2 W^* \,(T+1) + C_3 W^{*2} T^* \qquad (13)$$

where

$$C_1 = 2\frac{A_p}{A_m}\Big(K_p - \frac{A_p}{A_m}\,\beta\Big),$$

$$C_2 = -2\Big(\frac{A_p}{A_m}\Big)\beta, \text{ and}$$

$$C_3 = 2\frac{A_p}{A_m}\Big(\frac{1}{A^*} - \frac{A_m}{2A^*A_p}\,\beta - \frac{A_p}{A_m}\beta\Big).$$

As can be seen from equation (13),

$$\Delta P^* = F(W^*,T^*).$$

The additional dimensionless quantities listed below were used to correlate the static pressure distribution down the length of the mixing stack.

$PMS^* = \dfrac{\frac{PMS}{\rho_s}}{\frac{U_p^2}{2g_c}}$ a pressure coefficient which compares the pumping head $\frac{PMS}{\rho_s}$ for the secondary flow to the driving head $\frac{U_p^2}{2g_c}$ of the primary flow, where PMS = static pressure along the mixing stack length.

$\frac{D}{S}$ ratio of the axial distance from the primary nozzles to the mixing stack entrance, (D) to the primary dimension of the mixing stack, (S)

D. EXPERIMENTAL CORRELATION

It is desirable to make a direct comparison of prototype and model performance on a one-to-one basis so that the effects of changes in geometric parameters on eductor performance may be readily evaluated. The ratio of absolute temperatures is the only parameter which was not controlled during the model testing. Therefore a means of presenting the experimental data for a given geometric configuration in a form which results in a pseudo-independence of the dimensionless groupings P* and W* upon T* must be developed. From equation (13) a satisfactory correlation of P*, T*, and W* for all temperatures and flow rates takes the form

$$\frac{\Delta P^*}{T^*} = \phi(W^* T^{*n}) \qquad (14)$$

where the exponent n is determined to be equal to 0.44. The details of the determination of 0.44 as the correlating exponent for the geometric parameters of the models tested is given in Ref. 1. To obtain an eductor model's pumping characteristic curve, the experimental data is correlated and analyzed using equation (14), that is, P^*/T^* is plotted as a function of $W^*T^{*0.44}$. This correlation is used to predict the open to the environment operating point. Variations in the eductor model's geometry will change the appearance of the pumping characteristic curve and facilitate comparison of pumping ability between models. For ease of discussion, $W^*T^{*0.44}$ will henceforth be referred to as the pumping coefficient.

E. EDUCTOR SYSTEM

The multiple nozzle eductor systems studied are designed specifically for service onboard gas turbine powered ships. The model consisted of a single primary uptake, a single cluster of four primary nozzles of constant cross section, as pictured in Figures 6, 7, and 8, and a single mixing stack.

F. MODEL GEOMETRY

1. Mixing Stack with Symmetric Plug

The mixing stack with symmetric plug is picture in Figure 5a, 5b and 5c and dimensionally illustrated in Figure 4. The mixing stack of length to hydraulic diameter ratio (L/D_h) equal to 2.5 was constructed using an aspect ratio of 1.5. The mixing stack was manufactured from 0.635 cm (0.25 in.) thick plexiglas for the flat sides and 0.102 cm (0.04 in.) thick sheet aluminum which was formed to the plexiglas sheets and secured by 4-40 screws. The plug surface was manufactured from the same sheet aluminum and formed to the symmetric openings in the plexiglas. This material was also secured using 4-40 screws. All interior corners were sealed with silicone epoxy to insure leakage was a minimum. Additional material was glued around the entrance region to create a 1.25 cm (0.50 in.) radius. This allows for a smooth flow of secondary air into the mixing stack and prevents separation which could occur with a straight edge. An additional frame was attached to the exit region to prevent warpage. Pressure taps were located along the centerline of the flow paths and along the centerline of the plug. These locations are shown in Figures 4, 5a and 5b.

2. Mixing Stack with Ported and Shrouded Plug

The mixing stack with symmetric plug was modified using the porting and shroud concept of Lemke and Staehli [7] . This configuration is pictured in Figure 10 and dimensionally illustrated in Figure 9. The porting arrangement, shown in Figure 11b, was located at the low pressure region of the plug. These ports, along with the top half of the plug, were covered with a shroud. The shroud was manufactured using 0.127 cm (0.05 in.) thick sheet brass. To give the shroud support, spacers were placed between the shroud and plug. These spacers were manufactured using the same 0.127 cm (0.05 in.) sheet brass. Two additional functions were provided by the spacers. First, the spacers ensured that air drawn in by the ports impinged on the top of the plug. This was accomplished by placing baffles in the inlet of the shroud. Secondly, the spacers would act as heat transfer fins within the shroud. The spacer construction is dimensionally illustrated in Figure 11a.

G. ALIGNMENT

The alignment of the mixing stack with the primary air flow nozzles was accomplished using a level, a 30.48 cm (12.0 in.) rule graduated in 0.25 mm (0.01 in.) and a 45.72 x 30.48 cm (18.0 x 12.0 in.) square. The graduated rule and square were to establish the stand-off distance (D/S) and to center the primary flow nozzles within the entrance area of the mixing stack. The geometric alignment was checked for accuracy using pressure readings at symmetric points on the model. Additional verification was obtained by subsequent symmetric exit velocity profile measurements. The three axis mounting stand, pictured in Figure 17 and 18 allowed alignment adjustments to be performed easily.

III. EXPERIMENTAL APPARATUS

Air is supplied to the primary nozzles by means of a centrifugal compressor and associated ducting schematically illustrated in Figure 1. The mixing stack configuration being tested is placed inside an air plenum containing an airtight partition so that two separate air flows, secondary and tertiary, may be measured. The air plenum facilitates the accurate measurement of secondary and tertiary air flows by using ASME long radius flow nozzles.

A. PRIMARY AIR SYSTEM

The circled numbers found in this section refer to locations on Figure 1. The primary air ducting is constructed of 16-gage steel with 0.635 cm (0.25 in.) thick steel flanges. The ducting sections were assembled using 0.635 cm (0.25 in.) bolts with air drying silicone rubber seals between the flanges of adjacent sections. Entrance to the inlet ducting 1 is from the exterior of the building through a 91.44 cm (3.0 ft) square to a 30.48 cm (1.0 ft) square reducer, each side of which has the curvature of a quarter ellipse. A transition section 2 then changes the 30.48 cm (1.0 ft) square section to a 35.31 cm (13.90 in.) diameter circular section 3 . This circular section runs approximately 9.14 m (30 ft) to the centrifugal compressor inlet.

A standard ASME square edged orifice (4) is located 15 diameters downstream of the entrance reducer and 11 diameters upstream of the centrifugal compressor inlet, thus insuring stability of flow at both the orifice and compressor inlet. Piezometer rings (5) are located one diameter upstream and one-half diameter downstream of the orifice. The duct section also contains a thermocouple just downstream of the orifice. Primary flow is measured by means of the standard ASME square edged orifice designed to the specifications given in the ASME power test code [5]. The 17.53 cm (6.902 in.) diameter orifice used was constructed out of 304 stainless steel 0.635 cm (0.25 in.) thick. The inside diameter of the duct at the orifice is 35.31 cm (13.90 in.) which yields a beta ($\beta = d/D$) of 0.497. The orifice diameter was chosen to give the best performance in regard to pressure drop and pressure loss across the orifice for the primary air flow rate used (1.71 Kg/sec (3.77 lbm/sec)).

The centrifugal compressor (7) used to provide primary air to the system is a Spencer Turbo Compressor, catalogue number 25100-H, rated at 6000 cfm at 2.5 psi back pressure. The compressor is driven by a three phase, 440, volt, 100 horsepower motor.

A manually operated sliding plate variable orifice (6) was designed to constrict the flow symmetrically and facilitate fine control of the primary air flow. During operation, the butterfly valve (8) , located at the compressor's discharge, provided adequate regulation of primary air flow, eliminating the necessity of using the sliding plate valve. The sliding plate valve was positioned in the wide-open position for all data runs.

On the compressor discharge side, immediately downstream of the butterfly valve, is a round to square transition (9) followed by a 90 degree elbow (10) and a straight section of duct. All ducting to this point is considered part of the fixed primary air supply system. A transition section (12) is fitted to this last square section which reduces the duct cross section to a circular section 29.72 cm (11.7 in.) in diameter. This circular ducting tapers down to a diameter of 26.30 cm (11.15 in.) to provide the primary air inlet to the eductor system being tested. The transistion is located far enough upstream of the model to insure that the flow reaching the model is fully developed.

B. SECONDARY AIR PLENUM

The secondary air plenum, pictured in Figure 14, is constructed of 1.905 cm (0.75 in.) plywood and measures 1.22 m by 1.22 m by 1.88 m (4 ft by 4 ft by 6.17 ft). It serves as an enclosure that can contain all or only part of the eductor model and still allow the exit plane of the mixing stack to protrude. The purpose of the secondary air plenum is to serve as a boundary through which secondary air for the eductor system must flow. Long radius ASME flow nozzles, designed in accordance with ASME power test codes [5] and constructed of fiberglass, penetrate the secondary air plenum, thereby providing the sole means for metering the secondary air reaching the eductor. Appendix D of Ref. 1 outlines the design and construction of the secondary air flow nozzles. By measuring the temperature of the air entering the pressure differential across the ASME flow nozzles, the mass flow rate of secondary air can be determined. Flexibility is provided in measurement of the mass flow rate of secondary air by employing flow nozzles with three different throat diameters: 20.32 cm (8 in.), 10.16 cm (4 in.), and 5.08 cm (2 in.). By using a combination of flow nozzles, a wide variety of secondary cross sectional areas can be obtained.

A secondary air flow straightener, shown in Figures 1 and 3, consisting of a double screen is installed 1.22 m (4 ft) from the open end of the secondary air plenum, between the ASME long radius nozzles and the primary air flow nozzles. The purpose of the straightener is to reduce any swirl effect that could result when only a small secondary air flow area exists.

C. TERTIARY AIR PLENUM

The tertiary air plenum, pictured in Figures 14 and 15, is constructed of 1.90 cm (0.75 in.) plywood and measures 1.22 m by 1.22 m by 1.22 m (4 ft by 4 ft by 4 ft). It serves as an enclosure that completely surrounds the mixing stack and allows the exit and entrance regions to protrude. An airtight rubber diaphram type seal, schematically illustrated in Figure 3 and pictured in Figures 16 and 17, is located at each end of the enclosure. This allows measurement of a tertiary air flow independent of the secondary air flow. Tertiary air flow is measured with the use of long radius ASME flow nozzles designed in accordance with ASME test codes [5] and constructed of fiberglass. These nozzles are located so that they penetrate the air-tight tertiary air plenum, thereby providing the sole means for metering the tertiary air reaching the eductor. By measuring the temperature of the air entering and the pressure differential across the ASME flow nozzles, the mass flow rate of tertiary air can easily be obtained. Flexibility in measuring the tertiary flow is provided by employing different size flow nozzles: two of 20.32 cm (8 in.) throat diameter, three of 10.16 cm (4 in.) throat diameter, and two of 5.08 cm (2 in.) throat diameter. By using various combinations of these flow nozzles, a wide variety of tertiary cross section flow areas can be obtained.

The interior of the tertiary air plenum is pictured in Figure 14. The stand which holds the mixing stack can be seen mounted inside the plenum. This stand, Figures 18 and 19, provides three axis adjustments to the mixing stack for alignment purposes. Figure 16 shows the diaphragm air seal at the entrance to the mixing stack, and Figure 17 shows the diaphragm air seal at the exit plane of the mixing stack. As can be seen, removable ports were located in the exit plane door to allow for adjustments to the mixing stack and instrumentation without removing the diaphragms.

D. INSTRUMENTATION

Pressure instrumentation for measuring gage pressures is located inside the primary air uptakes just prior to the primary nozzles, inside the secondary air plenum, inside the tertiary air plenum, and at various points on the model. A variety of manometers, pictured in Figure 22, were used to indicate the pressure differentials. A schematic representation of the pressure measuring instrumentation is illustrated in Figures 20 and 21. Monitoring of each of the various pressures was facilitated by the use of a scanivalve and a multiple valve manifold. The scanivalve was used to select the pressure tap to be read, while the multiple valve manifold allowed selection of the optimum manometer for the pressure being recorded. A vent was included in the multiple valve manifold which provided a means of venting the manometers between pressure readings. The valve manifold provided a selection of a 15.24 cm (6.0 in.) inclined water manometer, a 5.08 cm (2.0 in.) inclined water manometer, and a 1.27 cm (0.5 in.) inclined oil manometer (specific gravity 0.827). In addition, the following dedicated manometers were used in the system: a 50.80 cm (20 in.) single column water manometer connected to the primary air flow just prior to the primary nozzles, a 127 cm (50 in.) U-tube water manometer with each leg connected to a piezometric ring on either side of the orifice plate in the air inlet duct, and a 2.54 cm (1.0 in.) inclined water manometer connected to the upstream piezometric ring.

Primary air temperatures, measured at the orifice outlet and just prior to the primary nozzles, are measured with copper-constantan thermocouples. The thermocouples are in assemblies manufactured by Honeywell under the trade name Megapak. Polyvinyl covered 20 gage copper-constantan extension wire is used to connect the thermocouples to a Newport Digital Pyrometer, model number 267, which provides a digital display of the measured temperature in degrees Fahrenheit. Secondary/tertiary ambient air temperature is measured with a mercury-glass thermometer and recorded in degrees Fahrenheit.

Velocity profiles at the mixing stack exit plane are obtained by using a pitot tube, pictured in Figure 23. The tube is affixed to a mounting template which allows accurate determination of the major axis, minor axis and diagonal positions and distances. Alignment pins allow changes in velocity traverse directions to be made fast and accurately. The pitot tube is used in conjunction with the 15.24 cm (6 in.) inclined water manometer for obtaining the velocity pressure head.

IV. EXPERIMENTAL METHOD

Evaluation of the eductor model requires the experimental determination of pressure differentials across the ASME long radius flow nozzles, temperatures of primary and induced air flows, internal mixing stack pressures, and mixing stack exit velocities. These experimentally determined quantities are then correlated and analyzed to obtain pumping coefficients, induced air flow rates, pressure distributions within the mixing stack, and mixing stack exit velocity profiles. The qualities of the eductor model are then evaluated to determine the model's effectiveness.

The following discussion addresses the individual qualities of the eductor model and how they were determined.

A. PUMPING COEFFICIENTS

The secondary pumping coefficient and the tertiary pumping coefficient provide the basis for analysis of the eductor model's pumping performance. Thus, changes in eductor model parameters which affect pumping can be noted by a change in pumping coefficient. The pumping coefficient(s) is desired at the operating point which is simulated by completely opening the air plenum(s) to the environment. This can not be conveniently measured. Therefore, the eductor's characteristics are determined, plotted, and then extrapolated to the operating point.

The pumping charactersitics of the eductor model are established by varying the associated induced air flow rate, either secondary or tertiary, from zero to its maximum measurable rate. This rate is determined by sequentially opening the ASME flow nozzles mounted in the appropriate plenum and recording the pressure drop across the nozzles. Values for nozzle cross sectional area, pressure drop, and induced air temperature are then used to calculate the dimensionless parameters $\Delta P^*/T^*$ and $W^*T^{*0.44}$ or $\Delta PT^*/TT^*$ and $WT^*TT^{*0.44}$ as described earlier in the Theory and Analysis section. The dimensionless parameters are then plotted as illustrated in Figure 28. Extrapolation of the pumping characteristics curve to intersect with the zero pressure/temperature coefficient abscissa locates the appropriate operating point coefficient of the model.

B. INDUCED AIR FLOWS

Two induced air flows are identified in this study: secondary and tertiary (cooling).

The secondary air flow indicates the amount of induced air passing through the secondary air plenum. The dimensionless quantity W* is the ratio of the secondary air flow rate to the primary air flow air.

The tertiary air flow indicates the amount of induced air passing through the tertiary air plenum. The dimensionless quantity WT* is the ratio of the tertiary air flow rate to the primary air flow rate.

C. PRESSURE DISTRIBUTIONS

The mixing stack axial static pressure was obtained using a series of pressure taps fixed to the mixing stack. These taps were placed in two axial rows, the rows being along the centerline of the flow. Along each row the taps were axially spaced in increments of one quarter primary dimension, (S). The exact location of the pressure taps is indicated on Figures 4, 5a and 5b. The stack pressure (PMS*) is plotted versus X/S to obtain a mixing stack pressure distribution, Figure 26.

The plug axial static pressure distribution was obtained using a similar series of pressure taps. The pressure taps were placed along the centerline of the plug and axially spaced at one quarter primary dimension,(S), increments. The plug pressure (PP*) is plotted versus X'/P to obtain a plug pressure distribution, Figure 27.

D. EXIT VELOCITY PROFILES

Velocity profiles at the mixing stack exit were calculated from the pressures measured using the pitot tube pictured in Figure 23. Since it was impractical to obtain a complete three-dimensional plot of velocities at the exit plate of the mixing stack, advantage was taken of the symmetry of the velocity profile resulting from the arrangement of the primary nozzles. Only three traverses were made. The first traverse passes vertically along the centerline down the mixing stack exit, the second traverse passes horizontally across the mixing stack. This traverse, along the centerline, passes across the top of the plug. The third traverse passes diagonally through the intersection of the centerlines. Figure 24 illustrates the orientation and identification of the three velocity traverses, while Figures 25a, 25b, and 25c show plots of the velocity traverses for the wide open secondary flow condition.

V. DISCUSSION OF EXPERIMENTAL RESULTS

Exhaust eductor systems designed for marine gas turbine applications must substantially cool exhaust gases, present an exterior stack surface temperature which will not give an easily detectable infrared signature, and effectively disburse exhaust gases. In order to evaluate the overall eductor model performance, four areas of performance were identified: the amount of secondary air flow induced by the primary air flow, referred to here as pumping; the degree of mixing of primary and induced air flows within the mixing stack system, referred to here as mixing; the amount of uptake back pressure impressed upon the turbine exhaust by the eductor system; and the amount of cooling air available to reduce the exterior stack temperature of the eductor system.

The eductor model and its modified configurations in this study were designed to shield the primary flow nozzles from an overhead view and to provide cooling to those surfaces which are visible from above.

A. QUANTATIVE MEASUREMENTS

Quantative measurement of the four areas of performance was required to evaluate the eductor model. Data on the model (base design and modifications) is located in the tables referenced in the description of each configuration.

The pumping, mixing, back pressure and cooling quantities of the model were evaluated in the following manner.

1. Pumping

The valves of the secondary and tertiary pumping coefficients were used to evaluate the pumping abilities of the eductor model. Values for the pumping coefficients were obtained from plots of experimental data using the correlations

$$\frac{\Delta P*}{T*} = \phi \; (W*T*^{0.44})$$

for the secondary pumping coefficient, and

$$\frac{\Delta PT*}{TT*} = \phi \; (WT*TT*^{0.44})$$

for the tertiary pumping coefficient. Tabulated values of the pumping coefficients for the eductor model configurations tested are included in the tables and plotted in the figures.

2. Mixing

Design changes to the mixing stack exit geometry made quantification of the model mixing quality more complex than it had been in the previous studies. A qualitative evaluation of mixing was made by comparison of exit velocity profiles in each traverse direction, Figure 24.

3. Uptake Back Pressure

The static uptake back pressure PU-PA was a value directly recorded from experimental data. To optimize turbine efficiency, this value should be kept as low as practical and still maintain an effective exhaust eductor system. Tabulated values of static uptake back pressure are referenced in the Tables.

4. Cooling

The value of the cooling pumping coefficinet $WT^*TT^{*0.44}$ is determined as described in the experimental method section. The values of the cooling pumping coefficient are proportional to the cooling air flow rate and indicate how well the eductor system is inducing cooling air.

B. SYSTEM EVALUATION

Initial model testing was conducted using an eductor model with a symmetric plug, a mixing stack length L/S of 3.0 and a standoff distance D/S of 0.5. The tests were conducted with a primary flow nozzle area to mixing stack area ratio of 3.0 these ratios of mixing stack length, standoff distance and primary nozzle to mixing stack area were used as a result of the previous study of Lemke and Staehli [7]. The results of this first set of eductor system tests are shown in Tables I and II and Figures 25, 26 and 27.

These tests showed three significant results. First, the uptake back pressure was significantly increased to 9.85 in. H_2O. This uptake back pressure is high and therefore would significantly impact on the turbine efficiency.

Secondly, the eductor model pumping coefficient was reduced by approximately thirty percent from the pumping coefficient of the model investigated by Lemke and Staehli, Figure 28. Finally, the velocity profiles in the horizontal and diagonal traverse directions showed high velocity peaks in the areas near the short side walls and in the corners. However, the vertical traverse shows no unique peak, Figures 25a, 25b, and 25c.

The first mixing stack modification attempted was that of porting and shrouding the interior (top-half) of the symmetric plug, Figures 9 and 10. The shrouding and baffle arrangement, shown in Figures 11 and 12, was designed such that the baffles ensured maximum air flow along the outer surface of the plug. The results of these tests are tabulated in Table III, IV, and V and Figures 30,32 and 35.

The tests of the model with porting and shrouding of the plug showed a further reduction in the model pumping coefficient, Figure 34. The resulting pumping coefficient had been lowered approximately thirty percent below the pumping coefficient of the model with no cooling air flow. A positive result of this modification was the reduction of the velocity peaks in the horizontal and diagonal traverses, Figures 30 and 31. This smoothing of the velocity profiles indicates that a more complete mixing is taking place in the mixing stack.

The final model configuration used the same porting and shroud arrangement with one exception. The row of ports farthest upstream on the plug (i.e. - closest to the mixing stack entrance) were covered to reduce the amount of cooling air flow. The same secondary conditions for each data run was maintained to ensure a means of comparing the results. The test results are tabulated in Tables VI, VII and VIII and Figures 30, 31, 32, and 35b.

The final configuration results showed only a slight improvement in the eductor system pumping coefficient from approximately 0.28 to 0.31, Figure 34. The cooling pumping coefficient was unchanged, Figure 33. The reduction of cooling air flow caused a small rise in velocity, but there was no significant change in the mixing. The uptake back pressure in both of the modified mixing stack configurations remained unchanged at 9.90 in H_2O.

Throughout the tests of the eductor model without cooling air, the axial pressure distribution was very nearly the same. Figures 26 and 32 show the plots of the axial pressure distriubtions of the mixing stack. Positive pressure readings occurred along the exterior curved surface of the mixing stack. However, a large negative pressure occurs at the junction of the curved surface and the rectangular mixing stack exit plane. No significant pressures were noted along the flat surface of the mixing stack.

The introduction of cooling air into the mixing stack caused changes in the axial pressure distributions. The pressures, both positive and negative, seen along the curved surface of the model without the cooling air flow were increased in magnitude with the addition of the cooling air. When one row of the ports was closed, these pressures were decreased by a small amount. The flat surfaces of the mixing stack also showed a change in the pressure distribution. Most significant of these changes was the shift from negative to positive pressures at distances of 0.5 to 1.75 (X/S) of the mixing stack. Figures 33a and 33b illustratively show these changes in the pressure distributions of the mixing stack with and without cooling air flow.

VI. CONCLUSIONS

This investigation studied the effects on the eductor system's overall performance of a mixing stack geometry which has a rectangular cross-section and employs a symmetric plug placed over the primary flow nozzles. A detailed description of the various eductor system model configurations is given in the experimental apparatus section. Interdependency of the model configurations was discussed in detail in the experimental results section. Only a summary of the conclusions resulting from this investigation are given here.

1. The base design model increased uptake back pressure and reduced the secondary pumping coefficient.
2. The ported and shrouded plug configuration further decreased the secondary pumping coefficient and improved mixing within the mixing stack.
3. Closing one row of ports did not significantly improve the secondary pumping coefficient while reducing the cooling pumping coefficient and had no effect on mixing.
4. The placement of the shroud and baffle arrangement directs flow to the top of the plug and provides cooling air to the plug surface.

VII. RECOMMENDATIONS

This study showed the effects on the eductor system's performance with a new geometric configuration. The results of the research are only the basis upon which future study and testing can be done. Recommendations for further investigation of this eductor system design are presented here.

1. Test the eductor model with the following configurations: the addition of an exterior shroud to the mixing stack, port the exterior of the mixing stack and the addition of a diffuser configuration to the exit plane.
2. Investigate the effects of changes in air flow area on the eductor system performance. Re-shaping of the plug and small area changes within the model may improve the pumping coefficient as well as reducing the uptake back pressure.

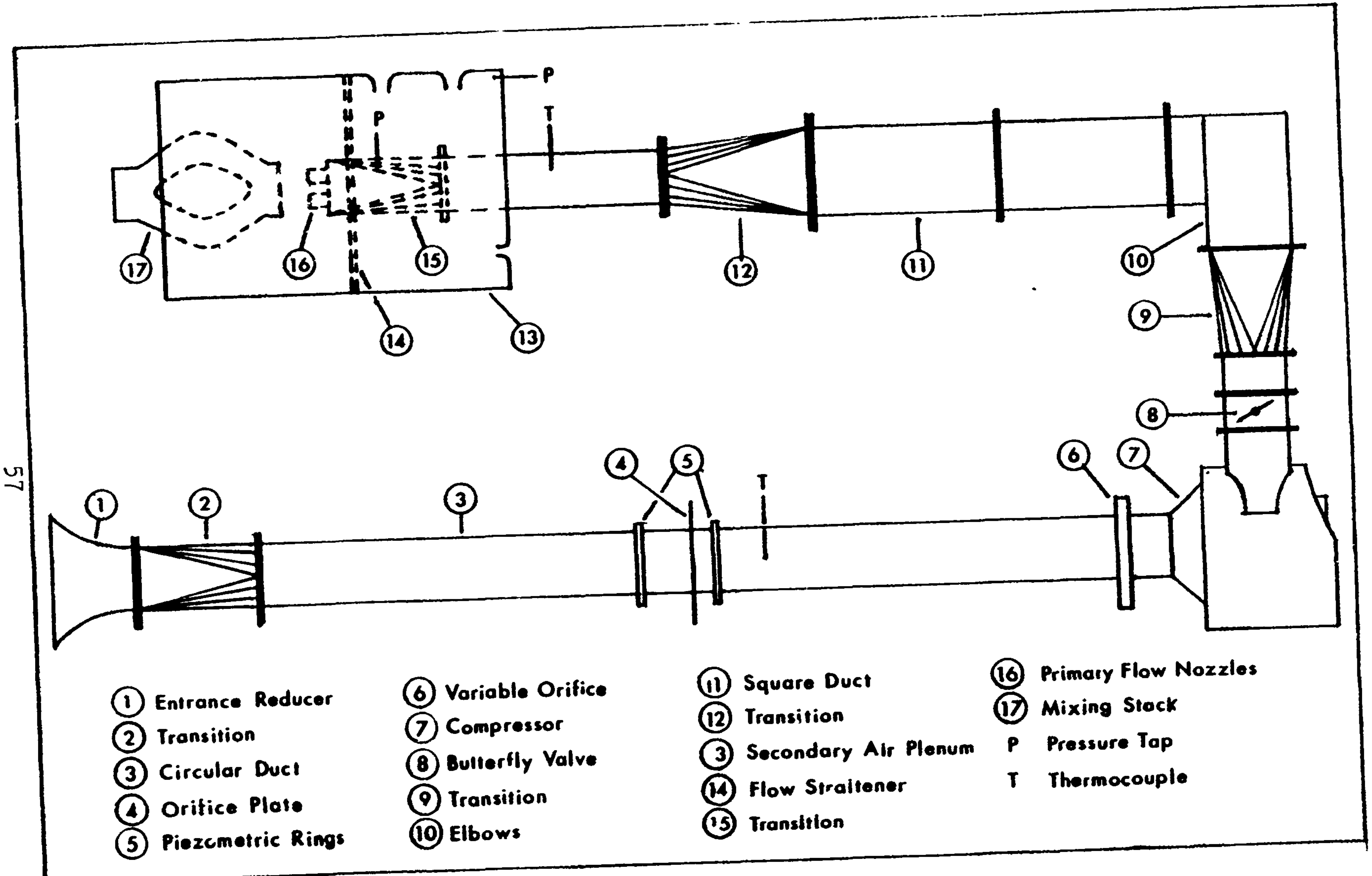

Figure 1. EDUCTOR MODEL TESTING FACILITY

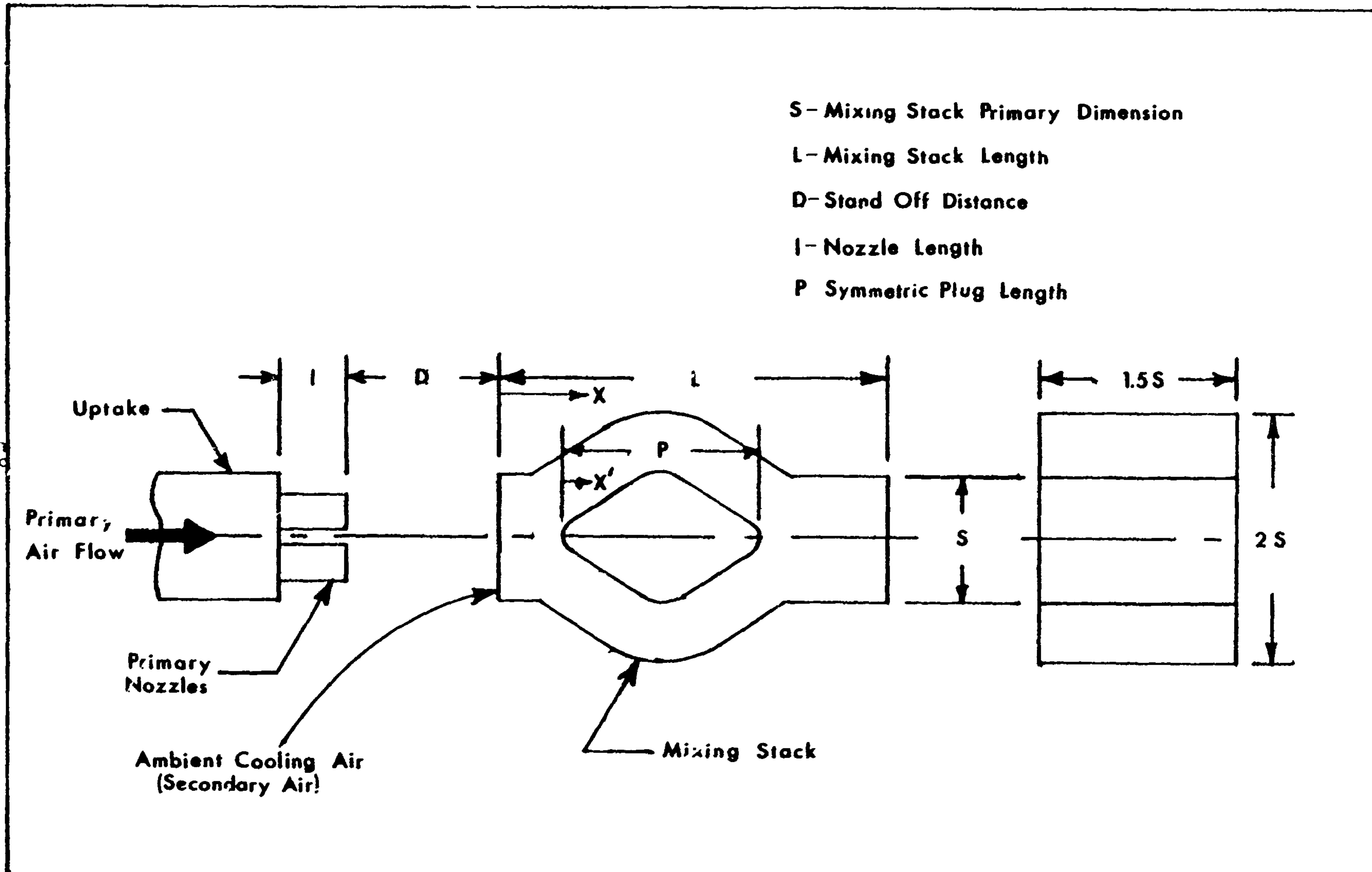

FIGURE 2. SCHEMATIC OF MIXING STACK AIR FLOW

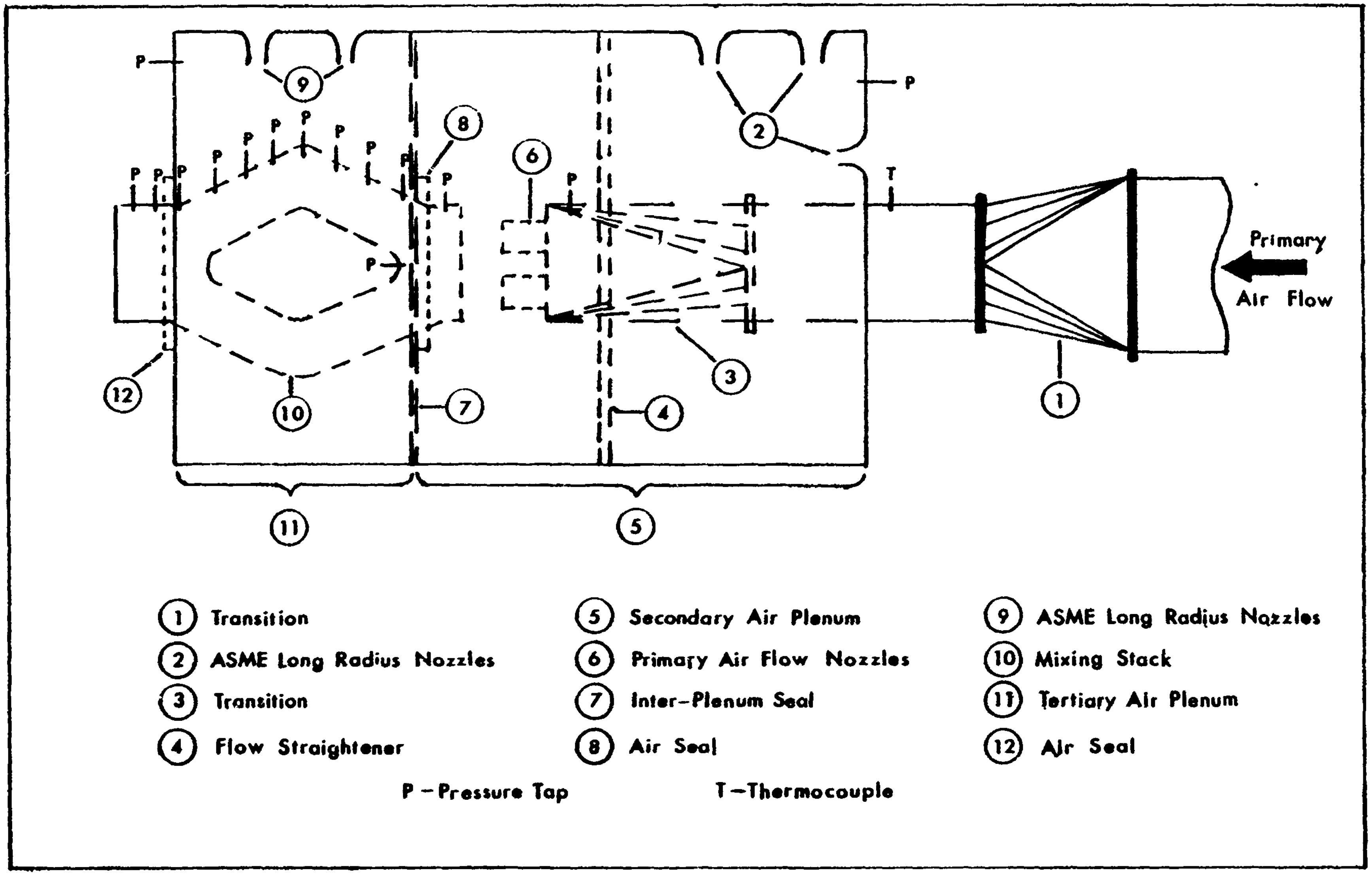

FIGURE 3. EDUCTOR MODEL TESTING FACILITY, SECONDARY AND TERTIARY AIR PLENUMS

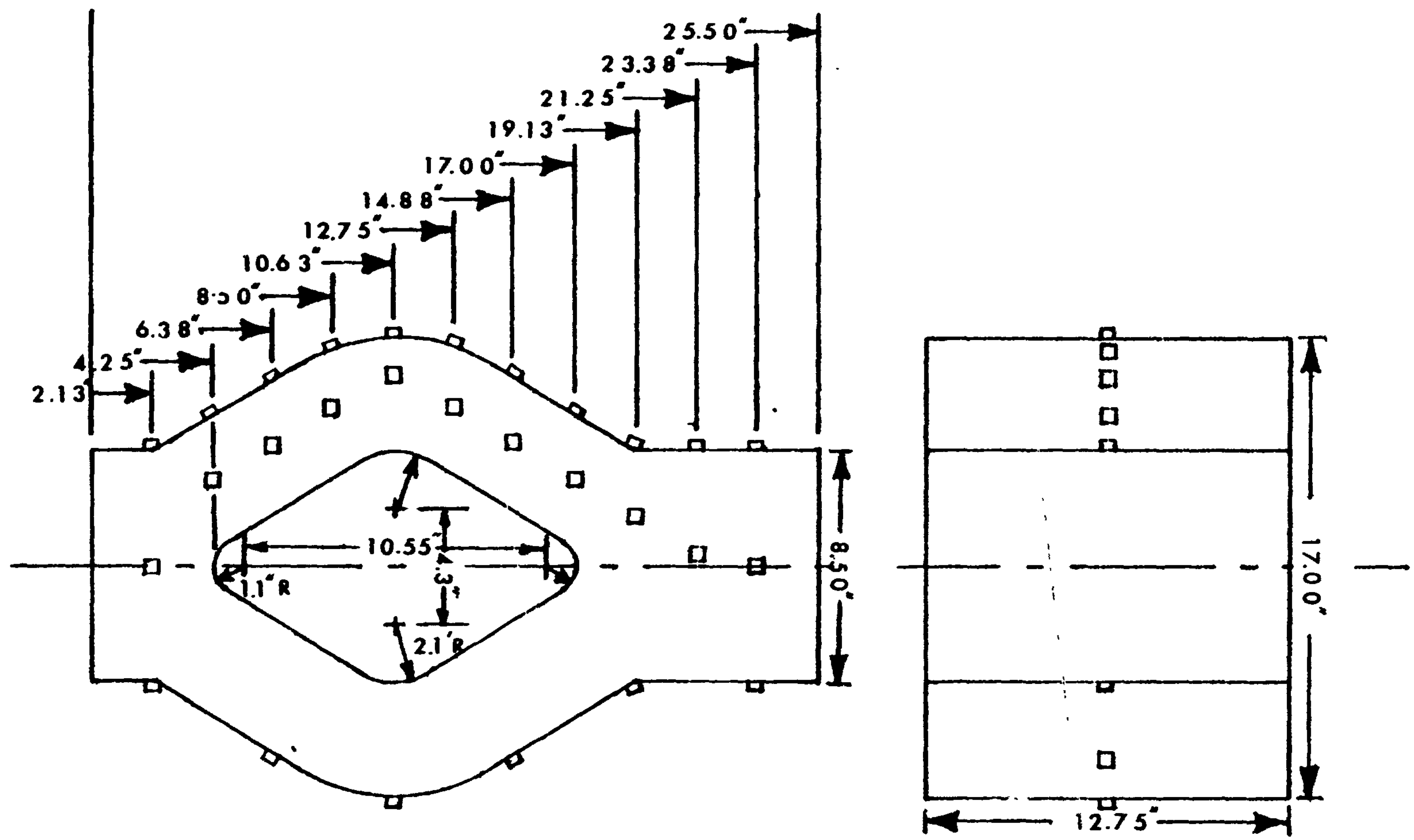

FIGURE 4. DIMENSIONAL ILLUSTRATION OF MIXING STACK WITH SYMMETRIC PLUG

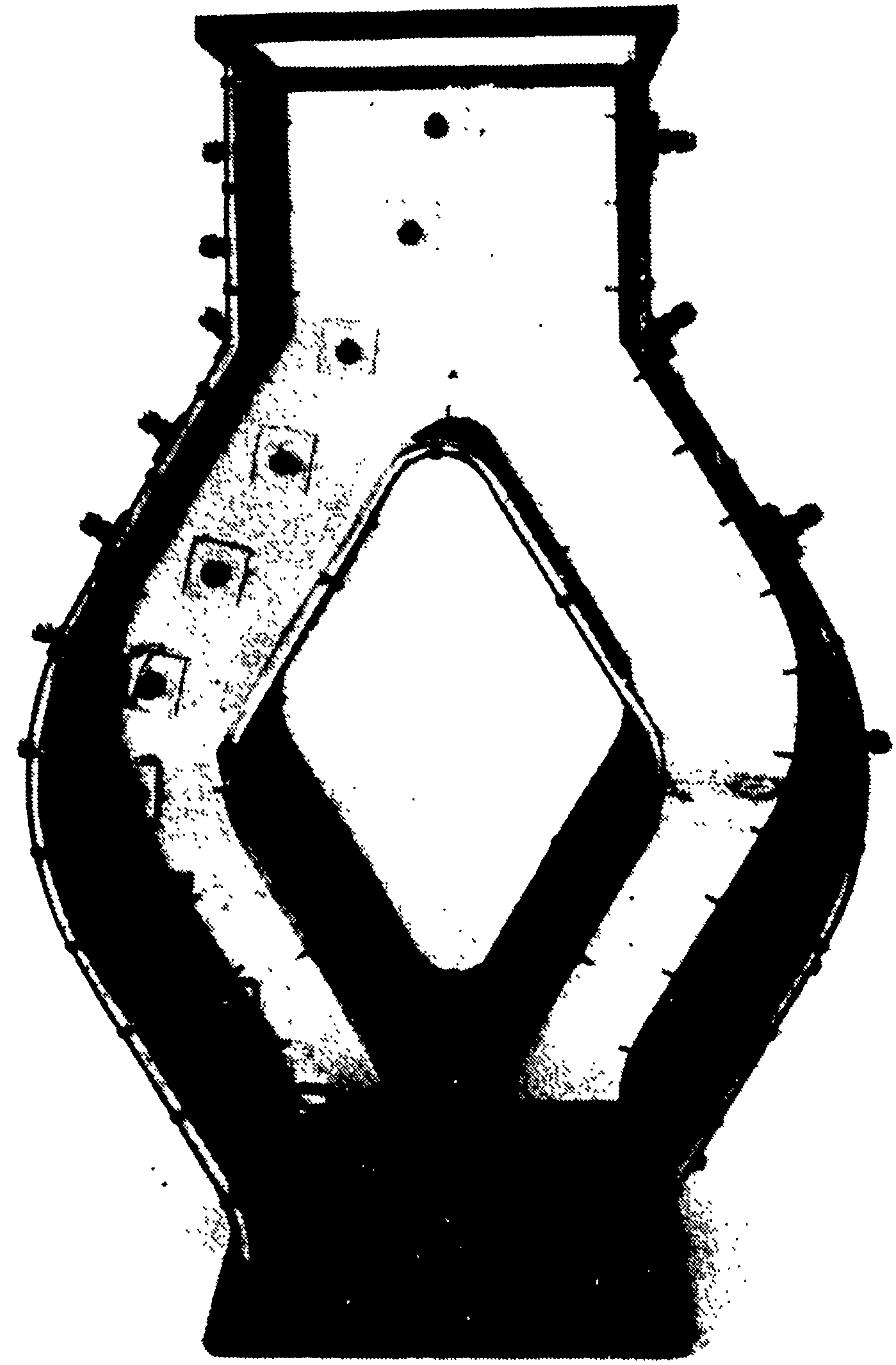

FIGURE 5a. MIXING STACK WITH SYMMETRIC PLUG (SIDE VIEW)

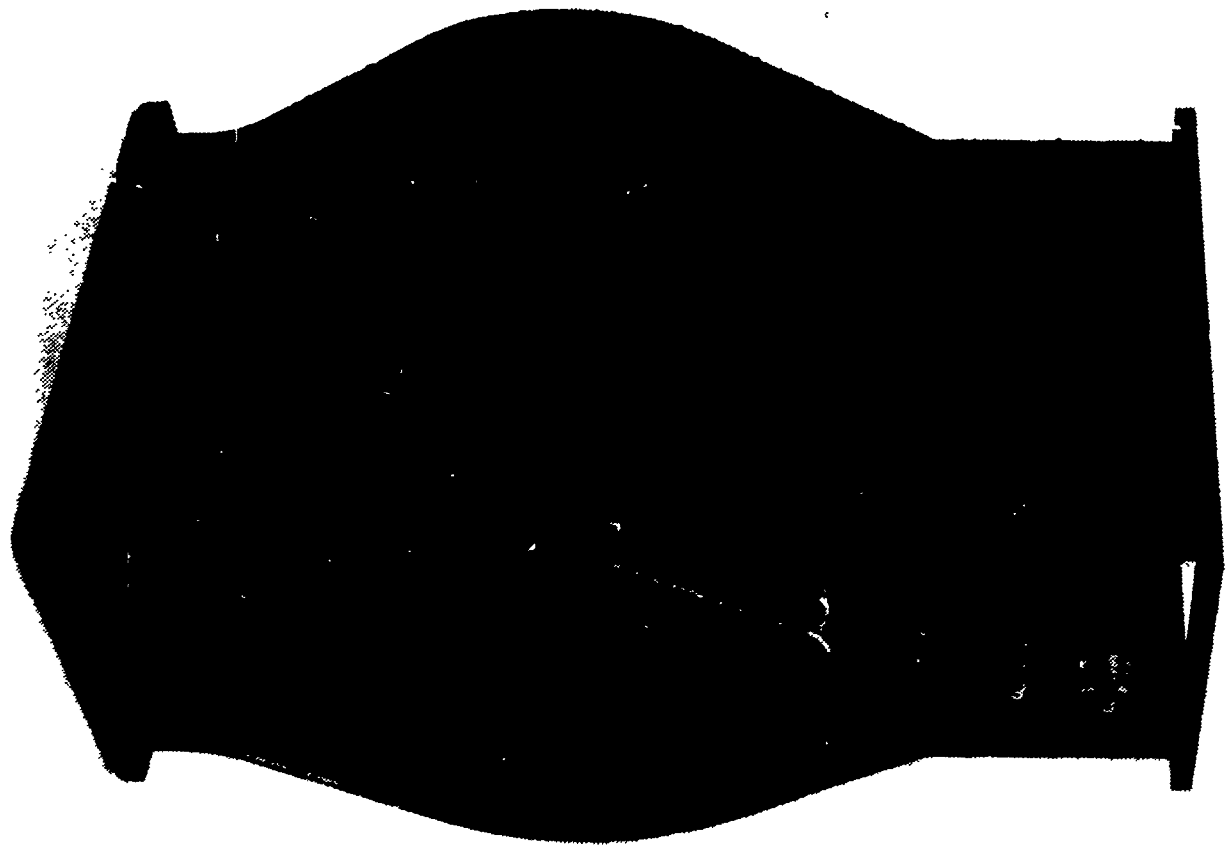

FIGURE 5b. MIXING STACK WITH SYMMETRIC PLUG (FRONT/SIDE VIEW)

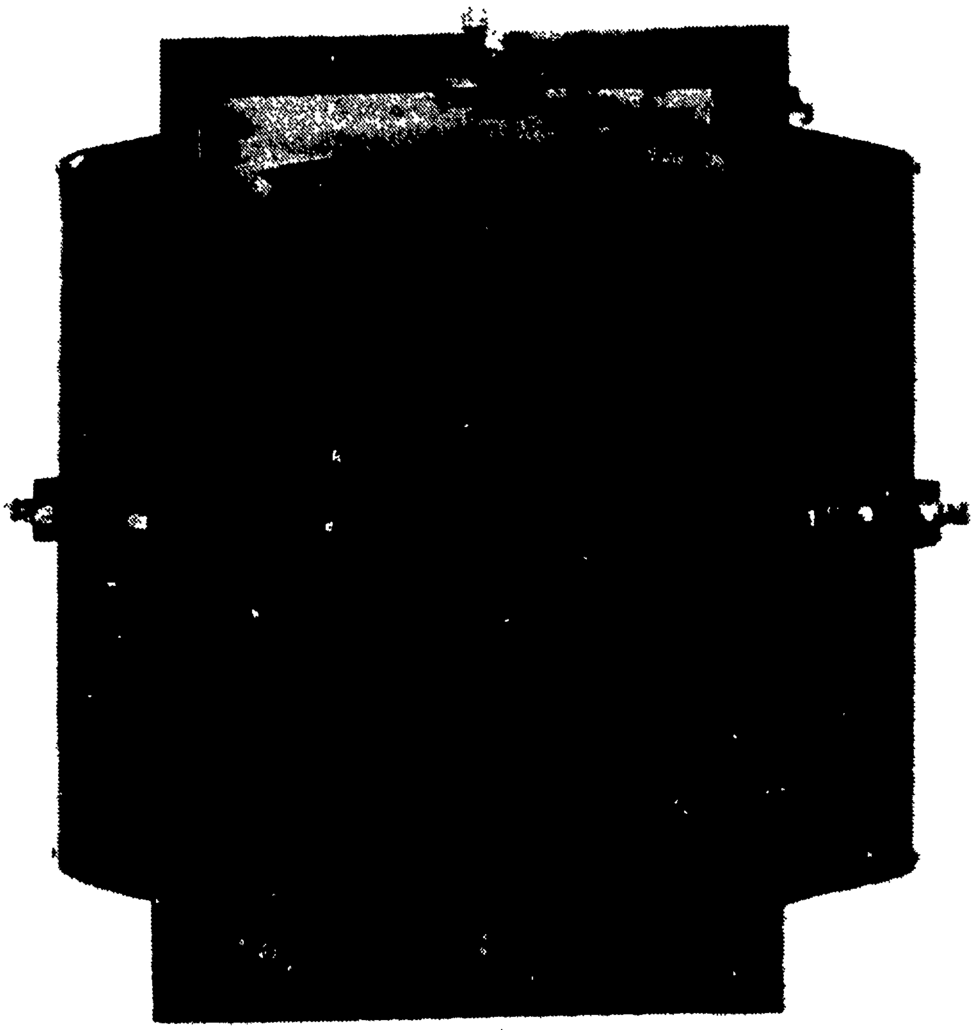

FIGURE 5c. MIXING STACK WITH SYMMETRIC PLUG (TOP VIEW)

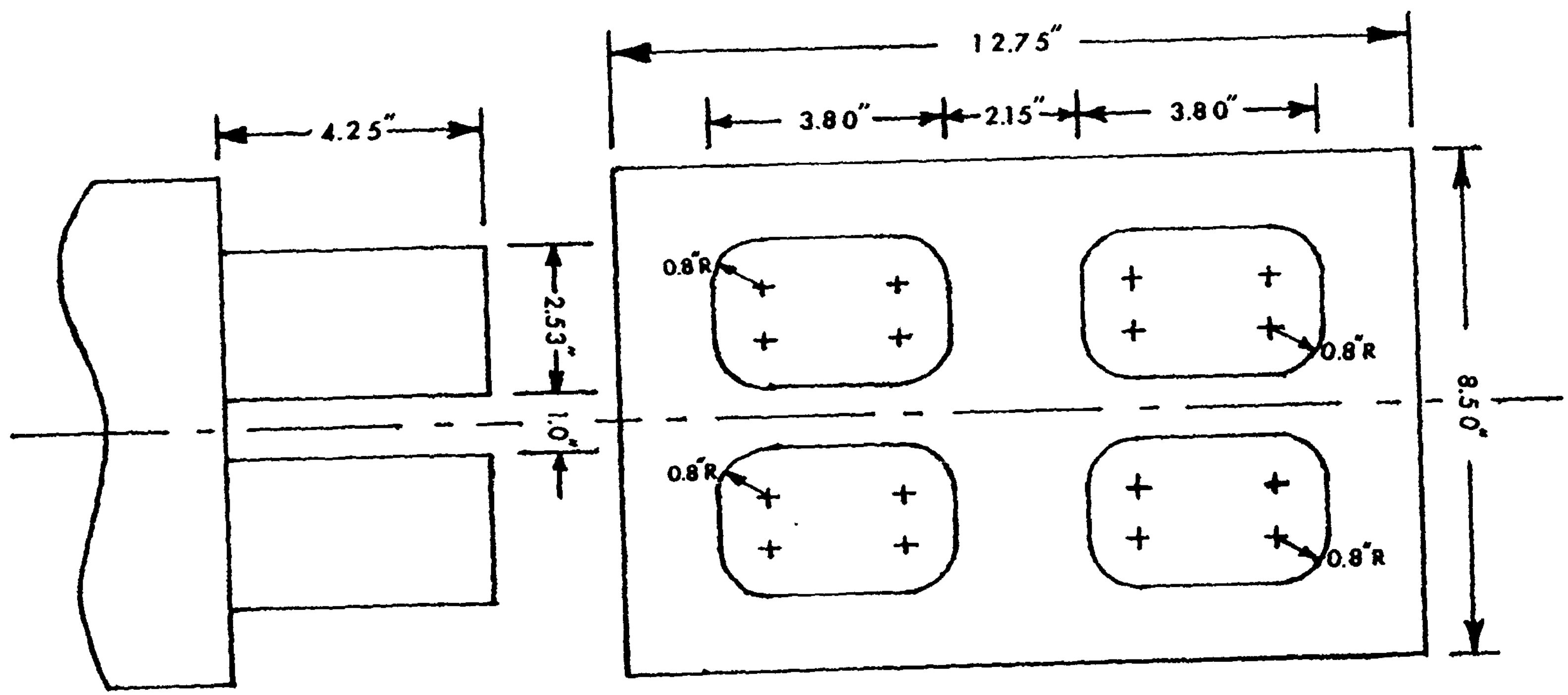

FIGURE 6. DIMENSIONAL ILLUSTRATION OF PRIMARY FLOW NOZZLES

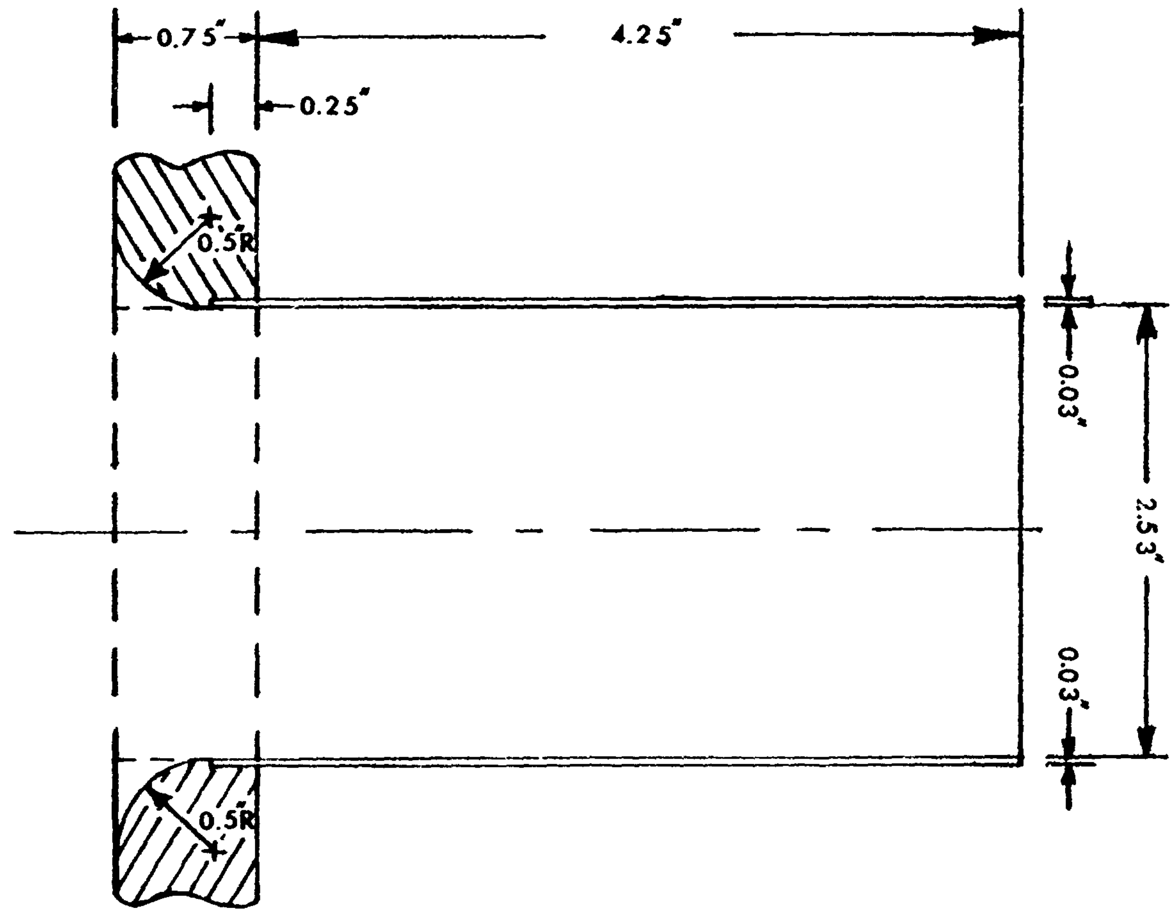

FIGURE 7. DIMENSIONAL ILLUSTRATION OF PRIMARY FLOW NOZZLES

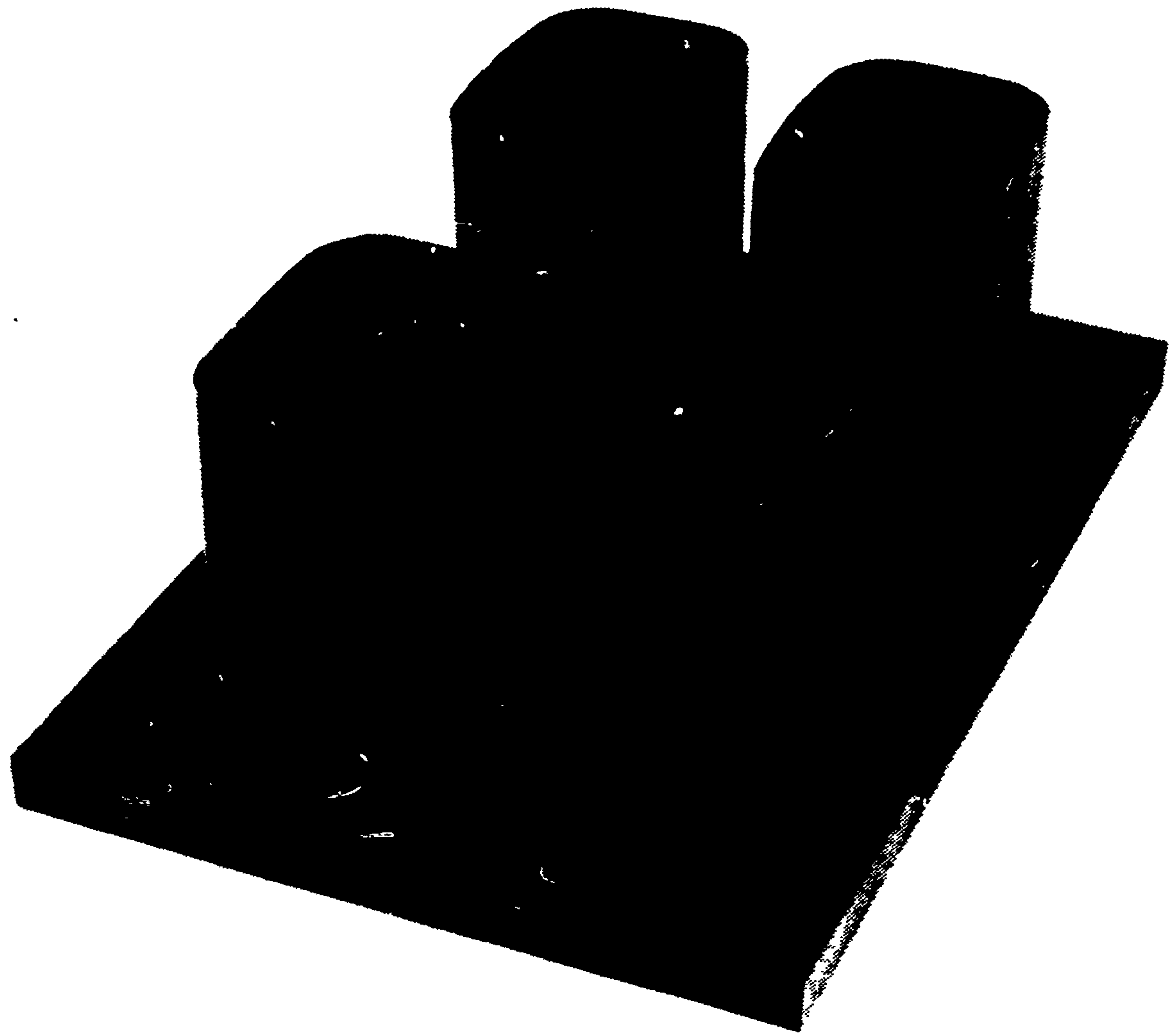

FIGURE 8. PRIMARY FLOW NOZZLES

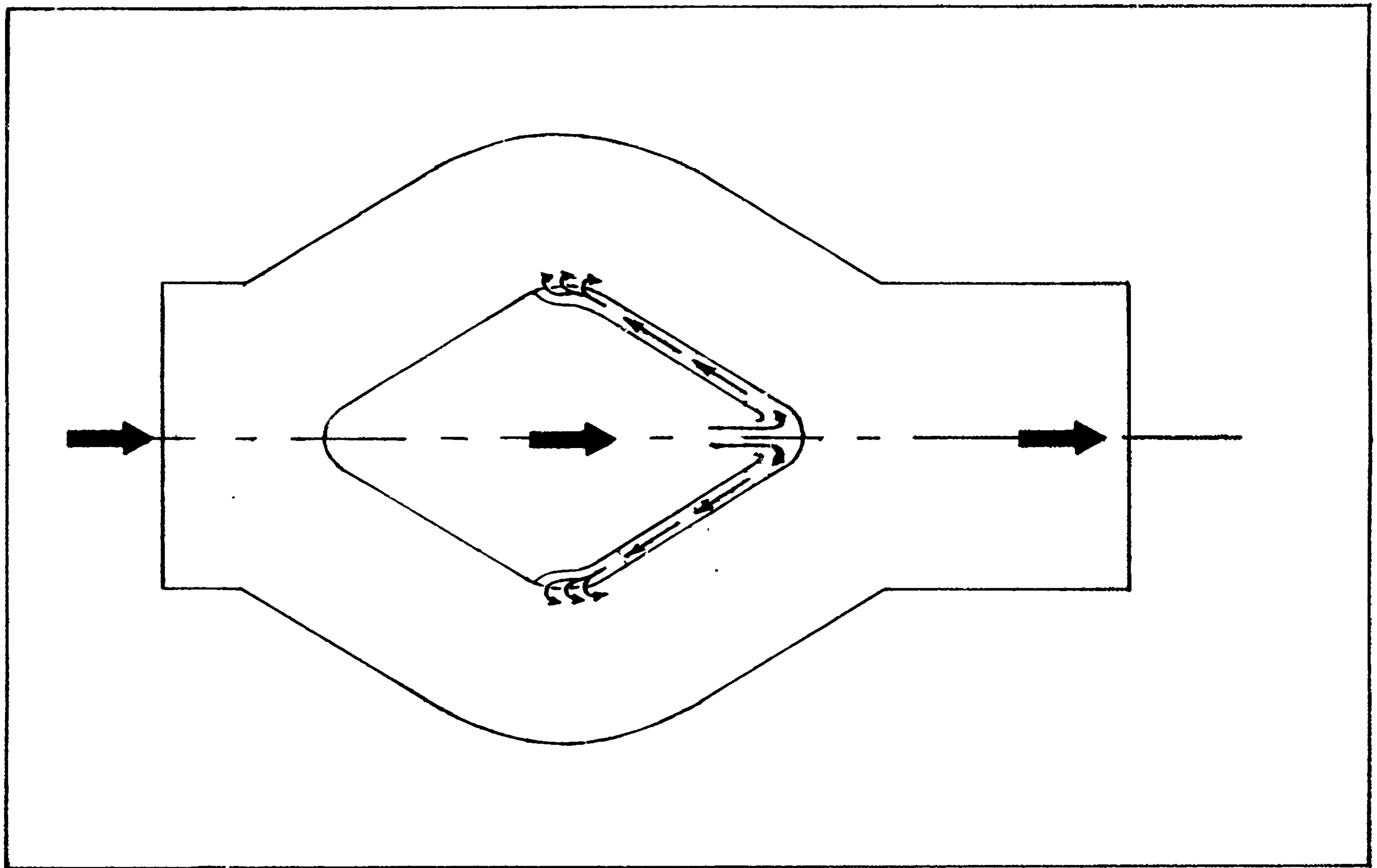

FIGURE 9. SCHEMATIC OF MIXING STACK WITH PORTED AND SHROUDED PLUG (CENTERLINE VIEW)

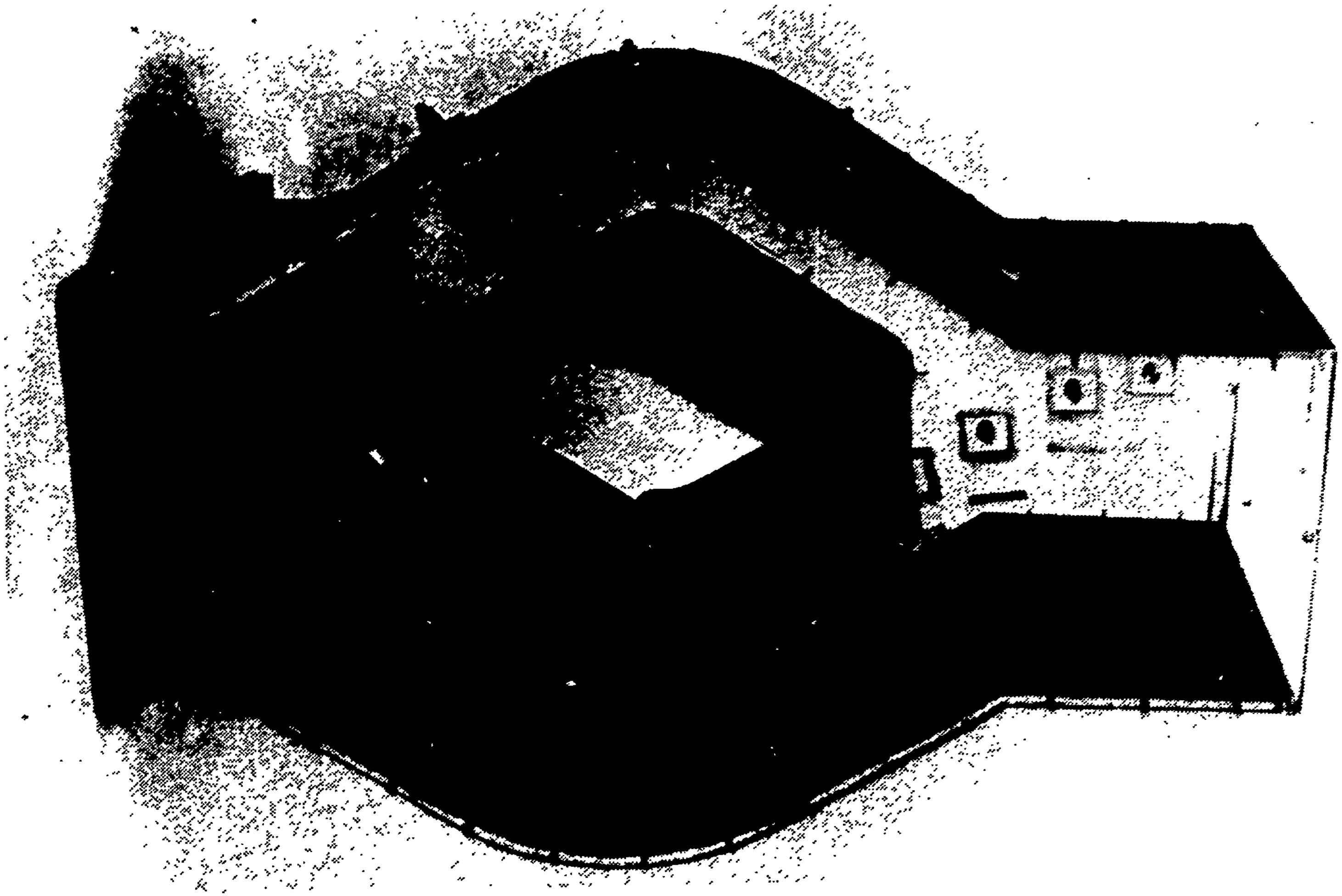

FIGURE 10. MIXING STACK WITH PORTED AND SHROUDED PLUG

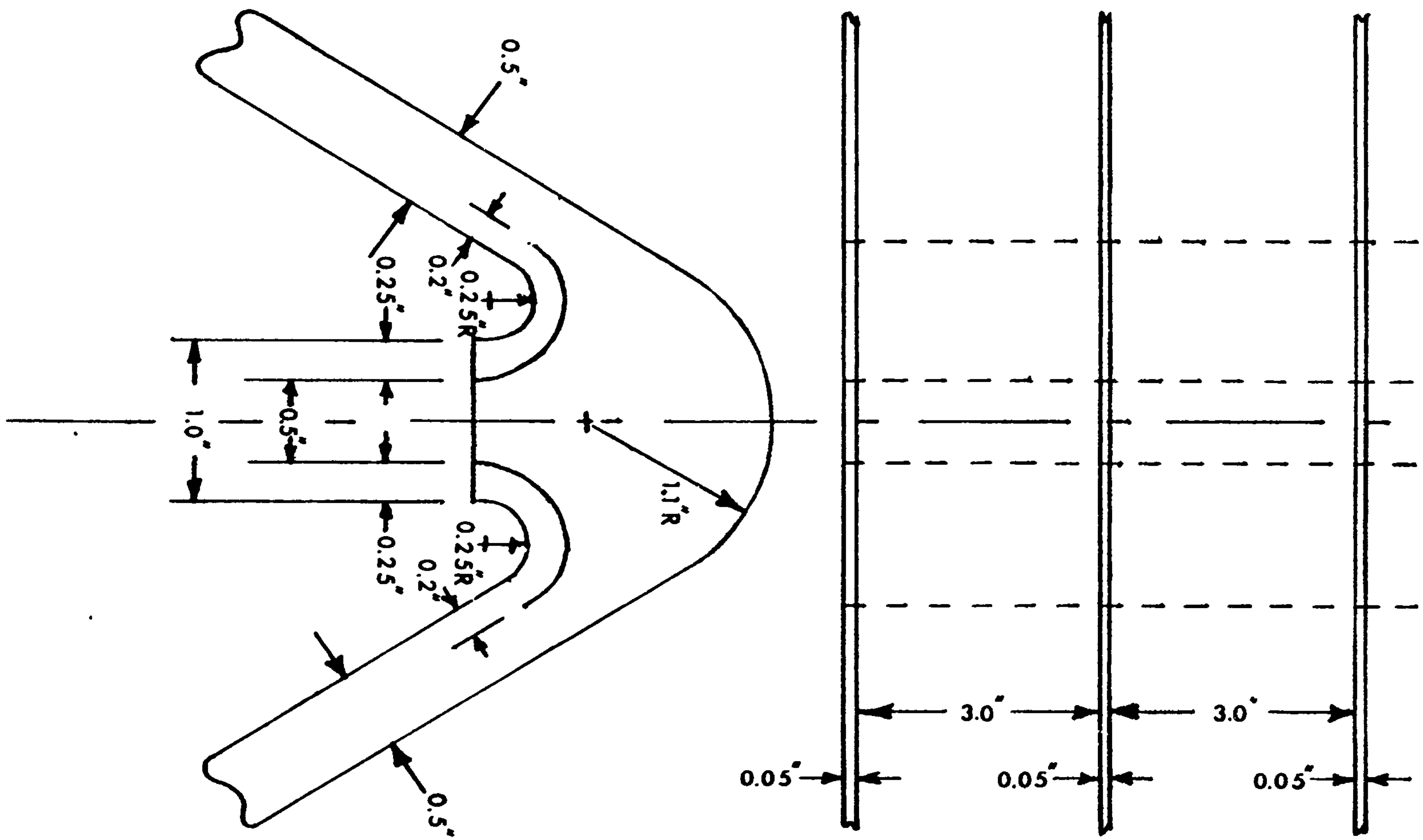

FIGURE 11a. DIMENSIONAL ILLUSTRATION OF SHROUDED SPACERS AND BAFFLES

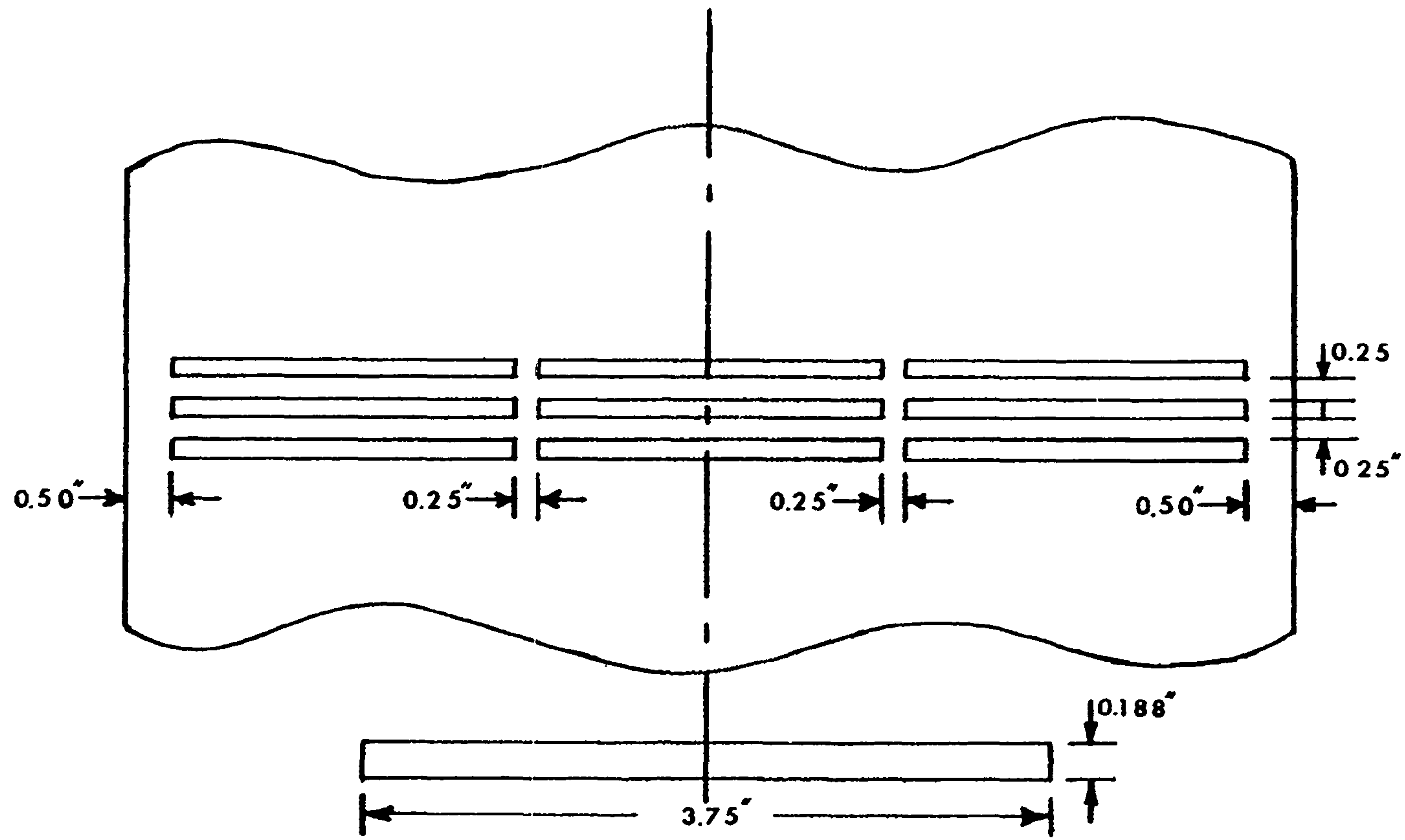

FIGURE 11b. DIMENSIONAL ILLUSTRATION OF PLUG PORTING ARRANGEMENT

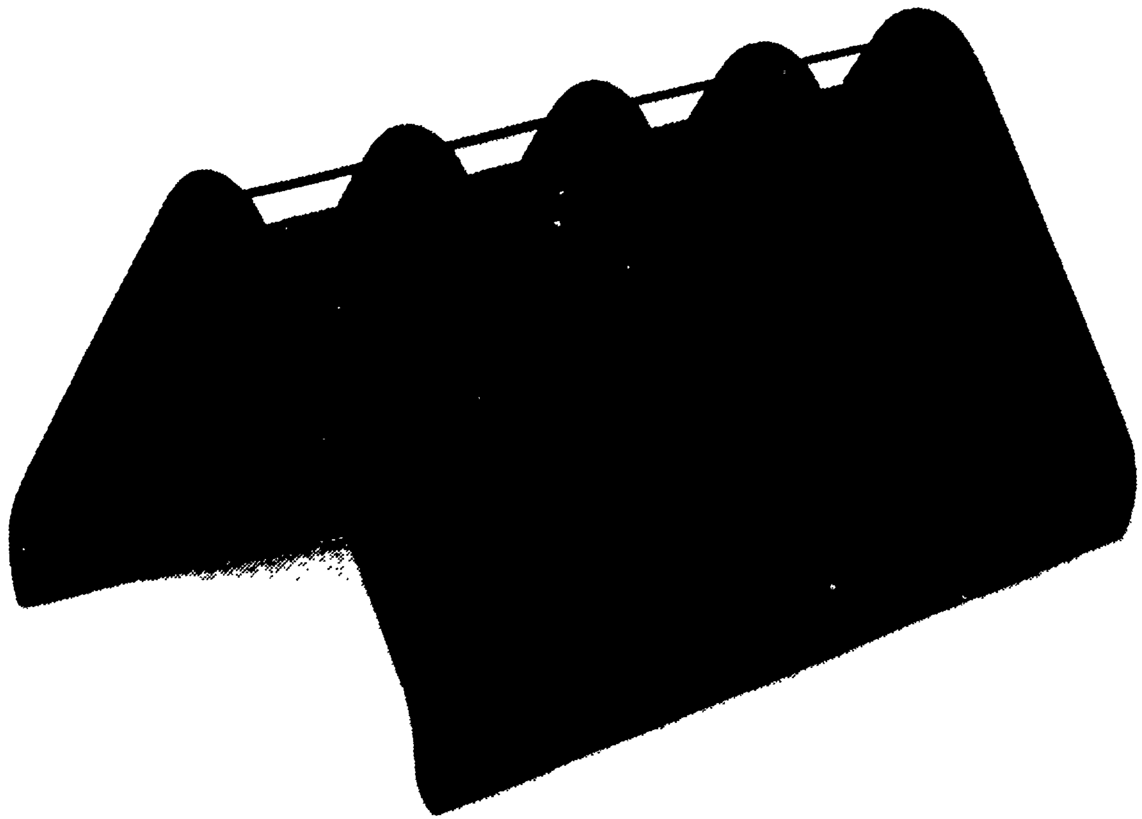

FIGURE 12a. SHROUD, BAFFLES AND SPACERS

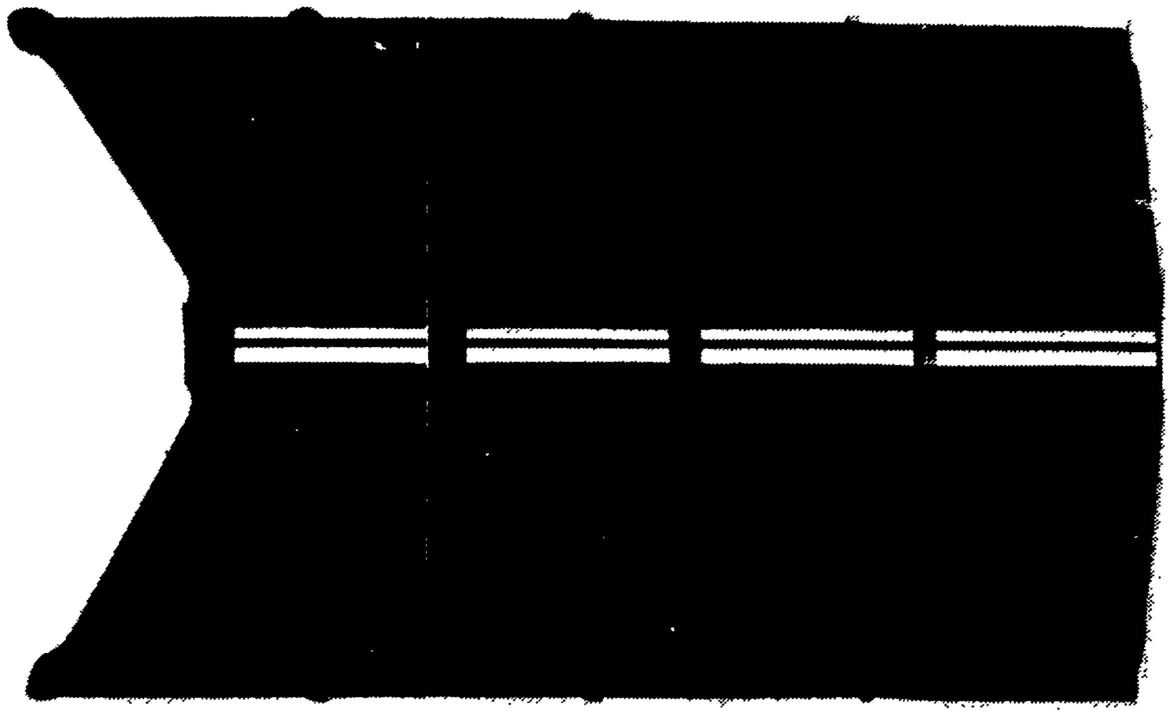

FIGURE 12b. SHROUD, BAFFLES AND SPACERS

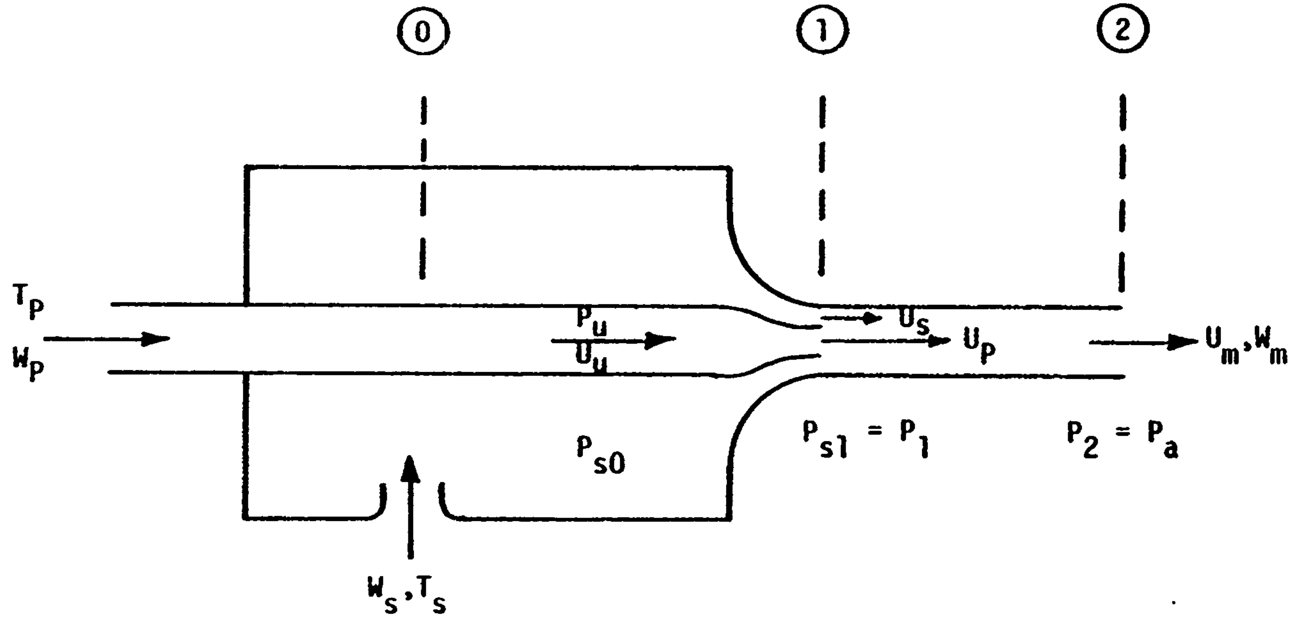

FIGURE 13. SIMPLE SINGLE NOZZLE EDUCTOR SYSTEM

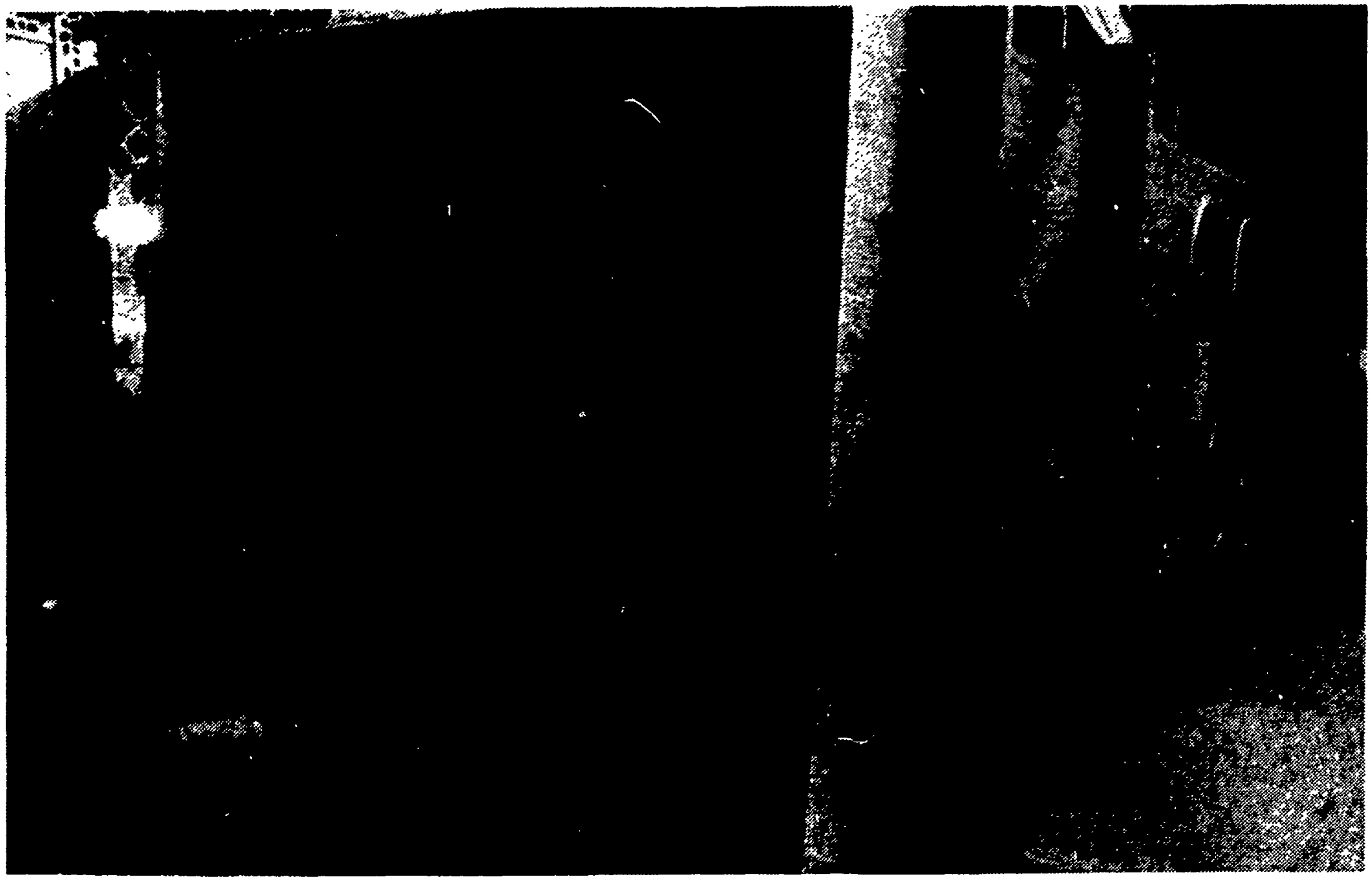

FIGURE 14. SECONDARY AND TERTIARY AIR PLENUMS

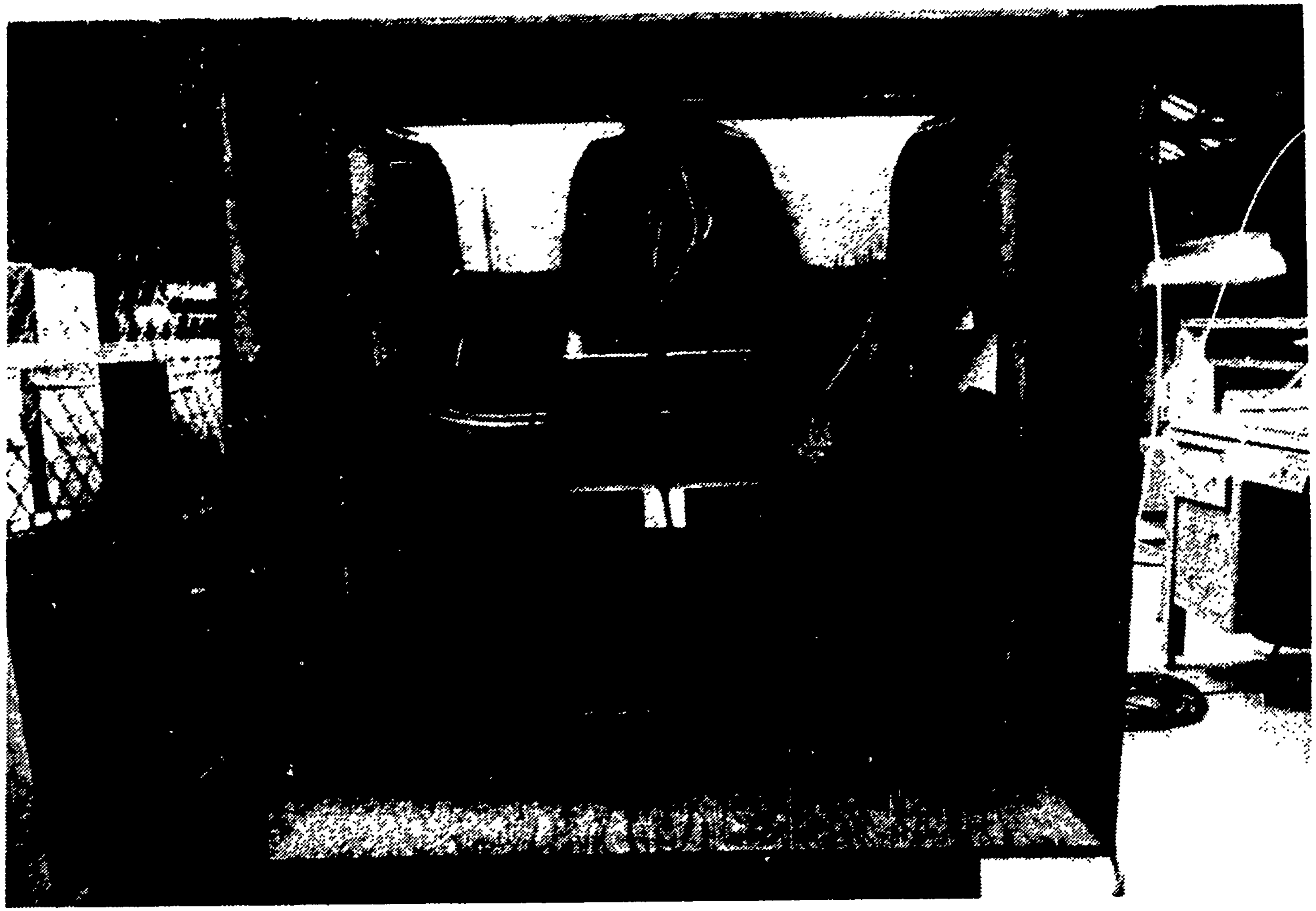

FIGURE 15. TERTIARY AIR PLENUM

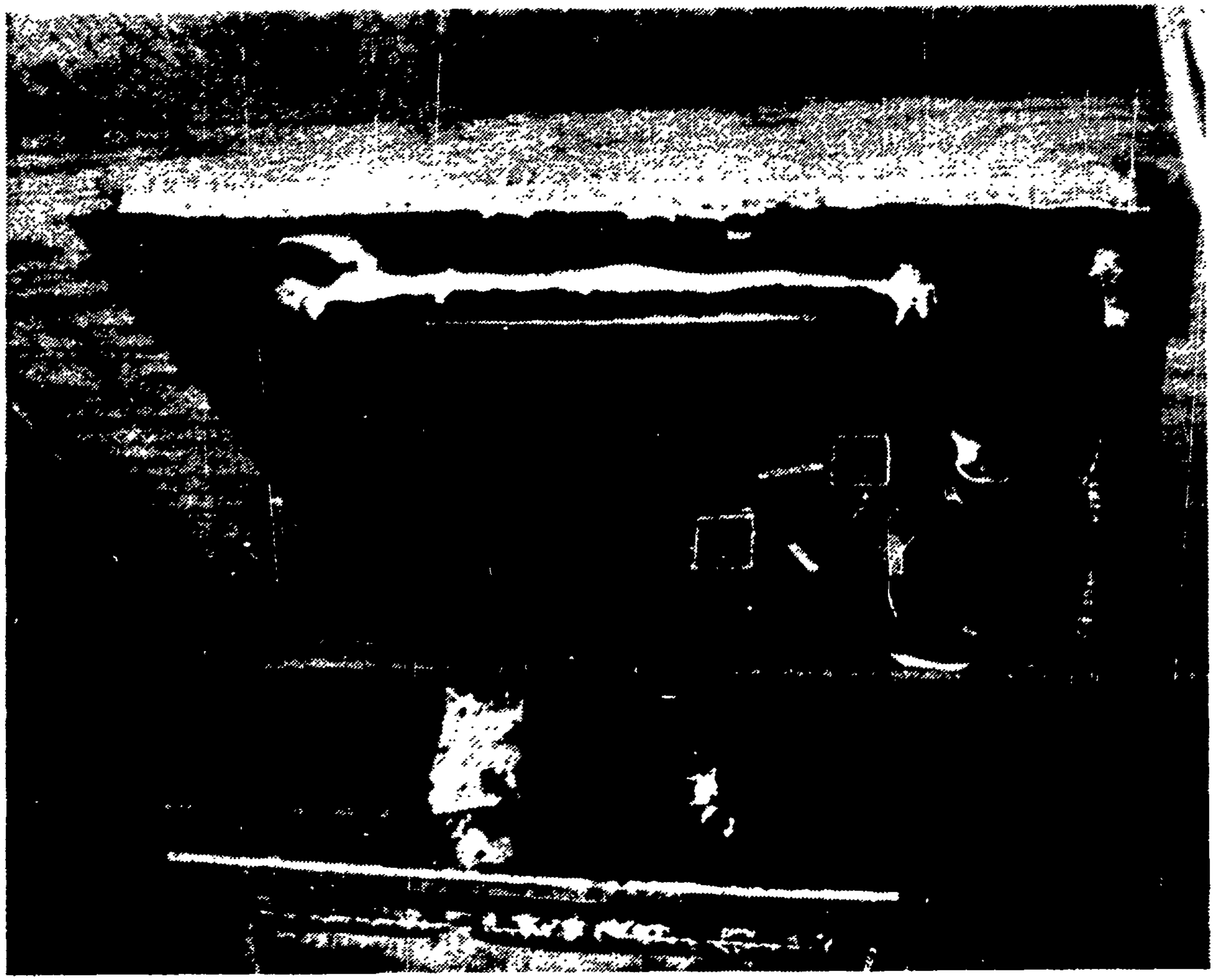

FIGURE 16. MIXING STACK ENTRANCE WITH AIR SEAL

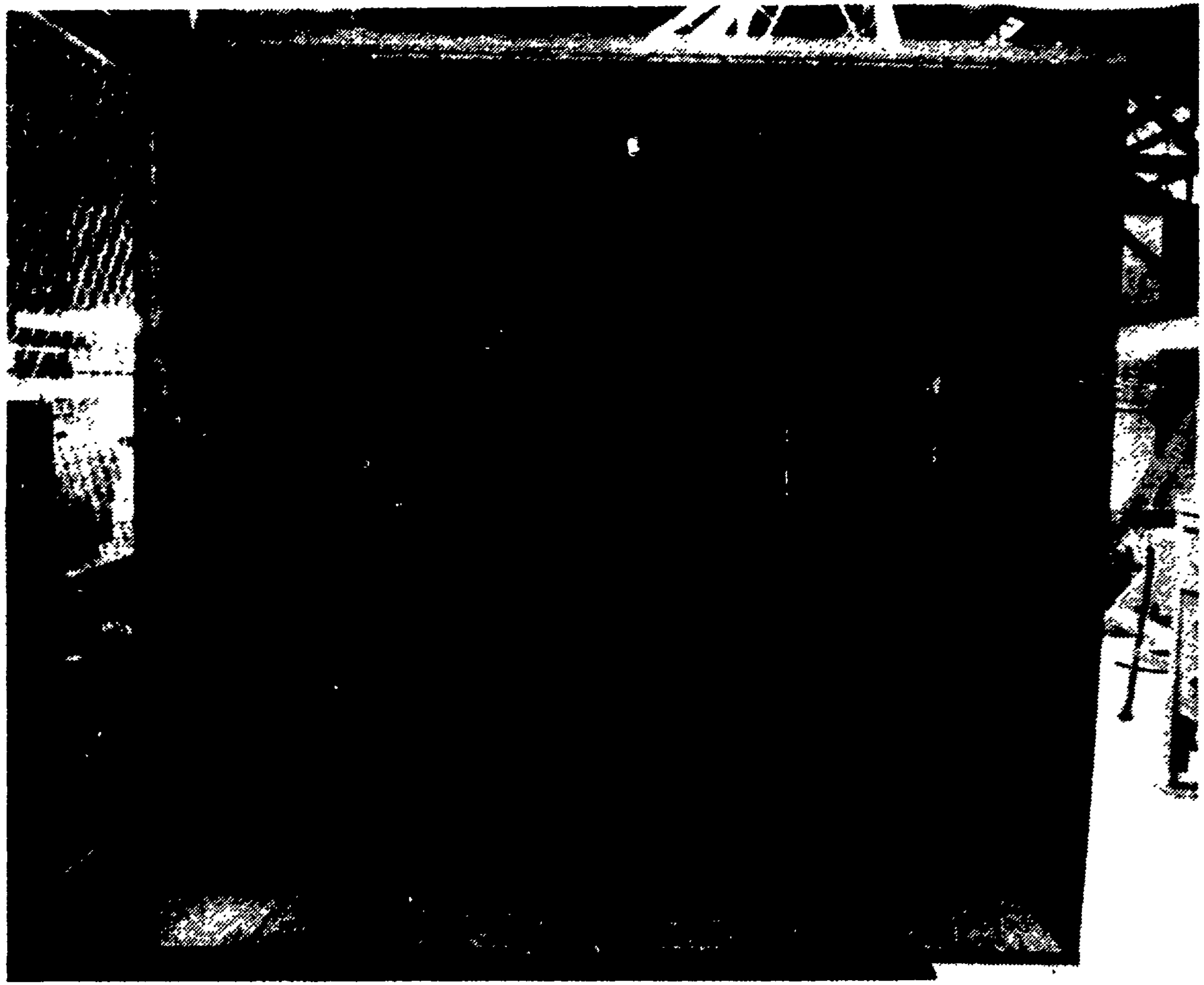

FIGURE 17. MIXING STACK EXIT WITH AIR SEAL

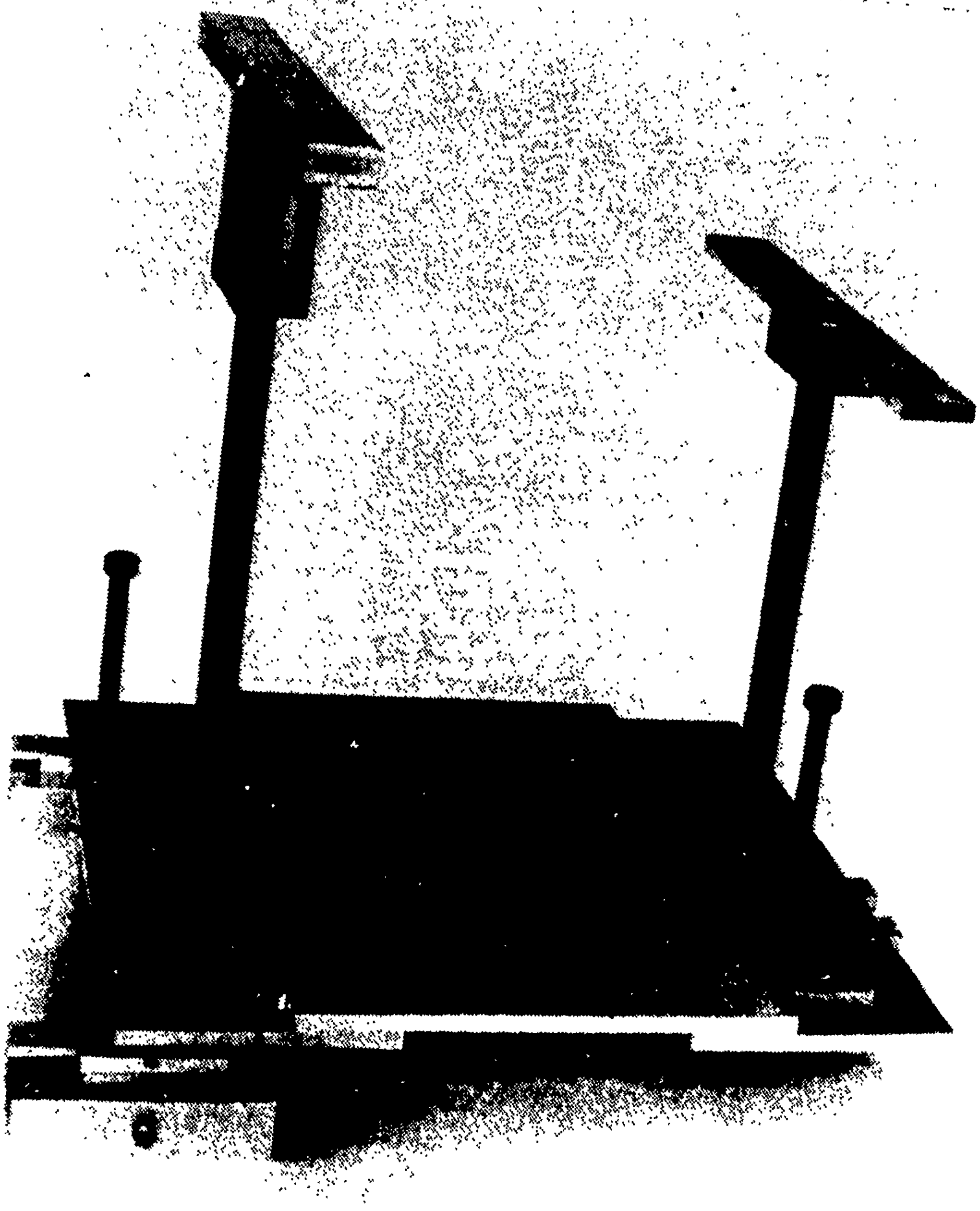

FIGURE 18. MIXING STACK MOUNTING STAND

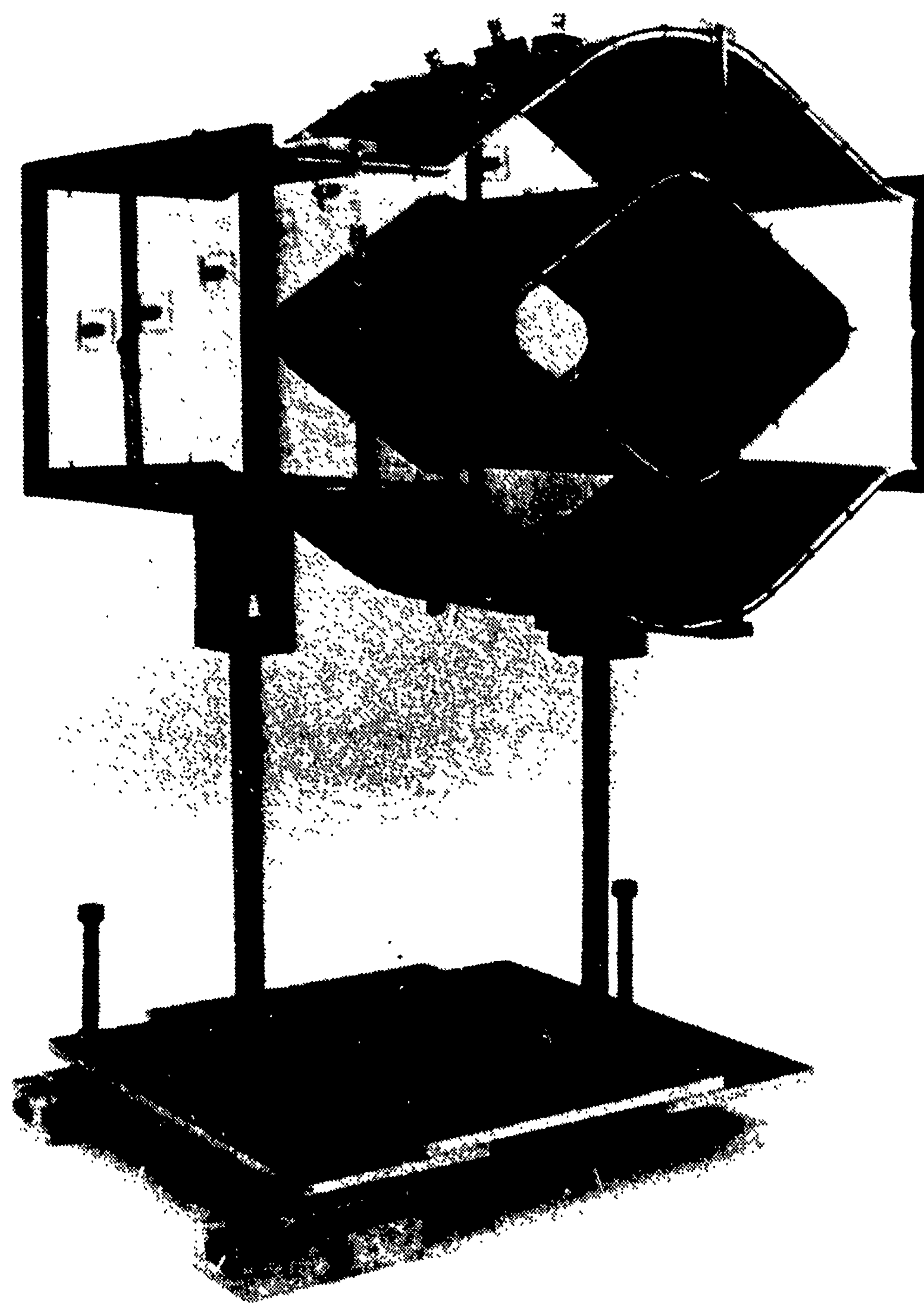

FIGURE 19. MIXING STACK AND MIXING STACK MOUNTING STAND

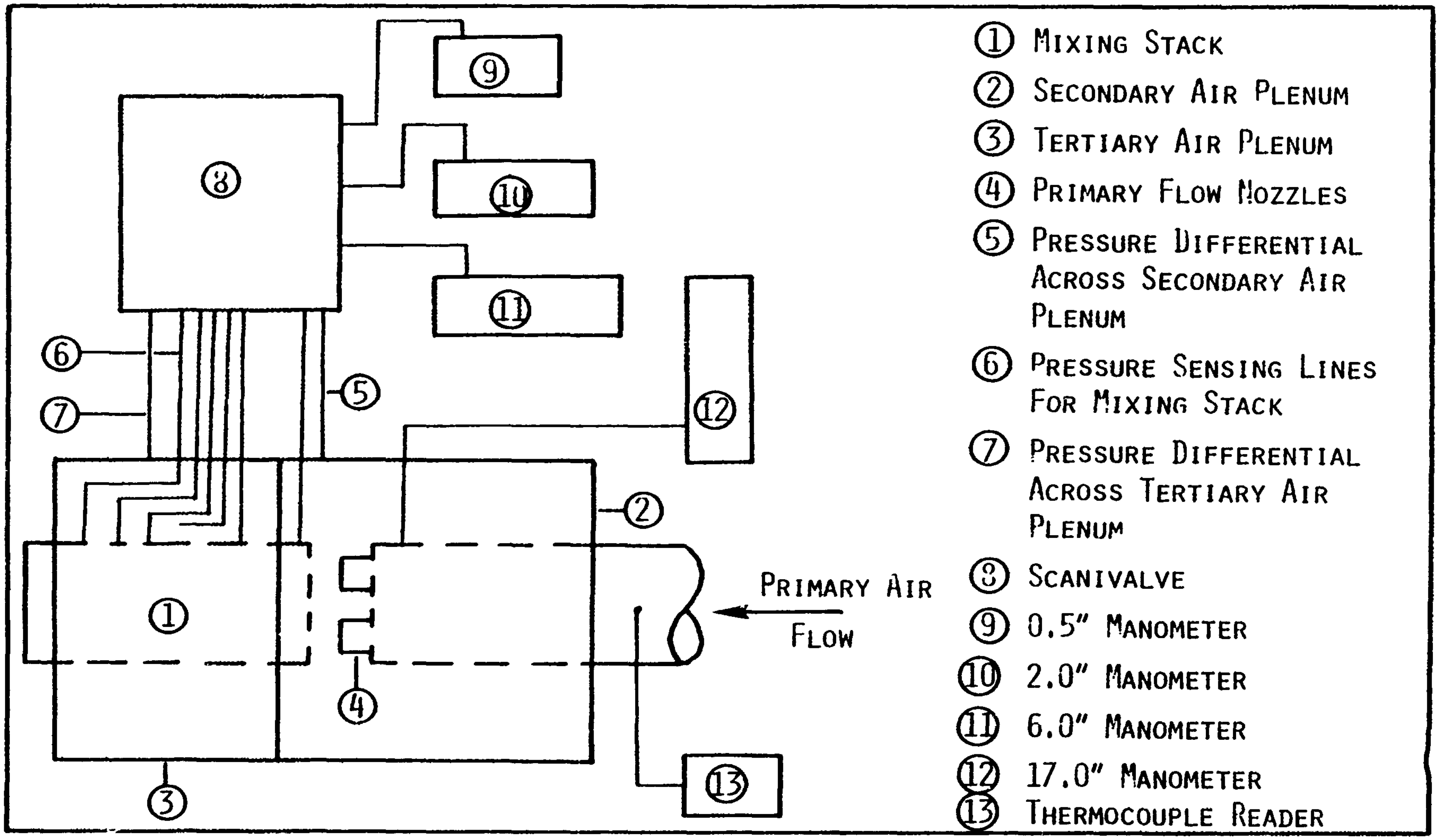

FIGURE 20. SCHEMATIC OF INSTRUMENTATION

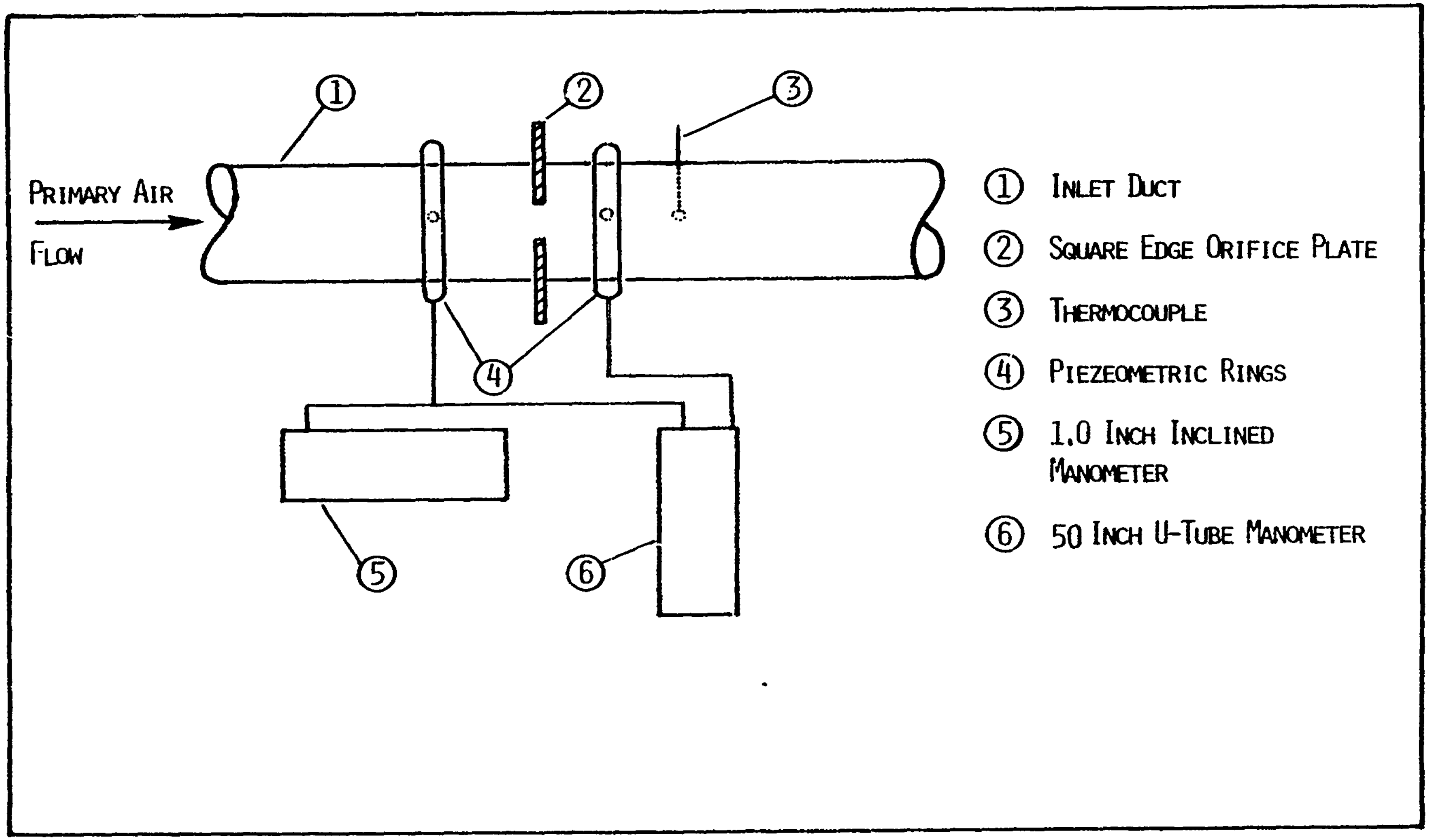

FIGURE 21. SCHEMATIC OF INSTRUMENTATION FOR PRIMARY AIR FLOW MEASUREMENT

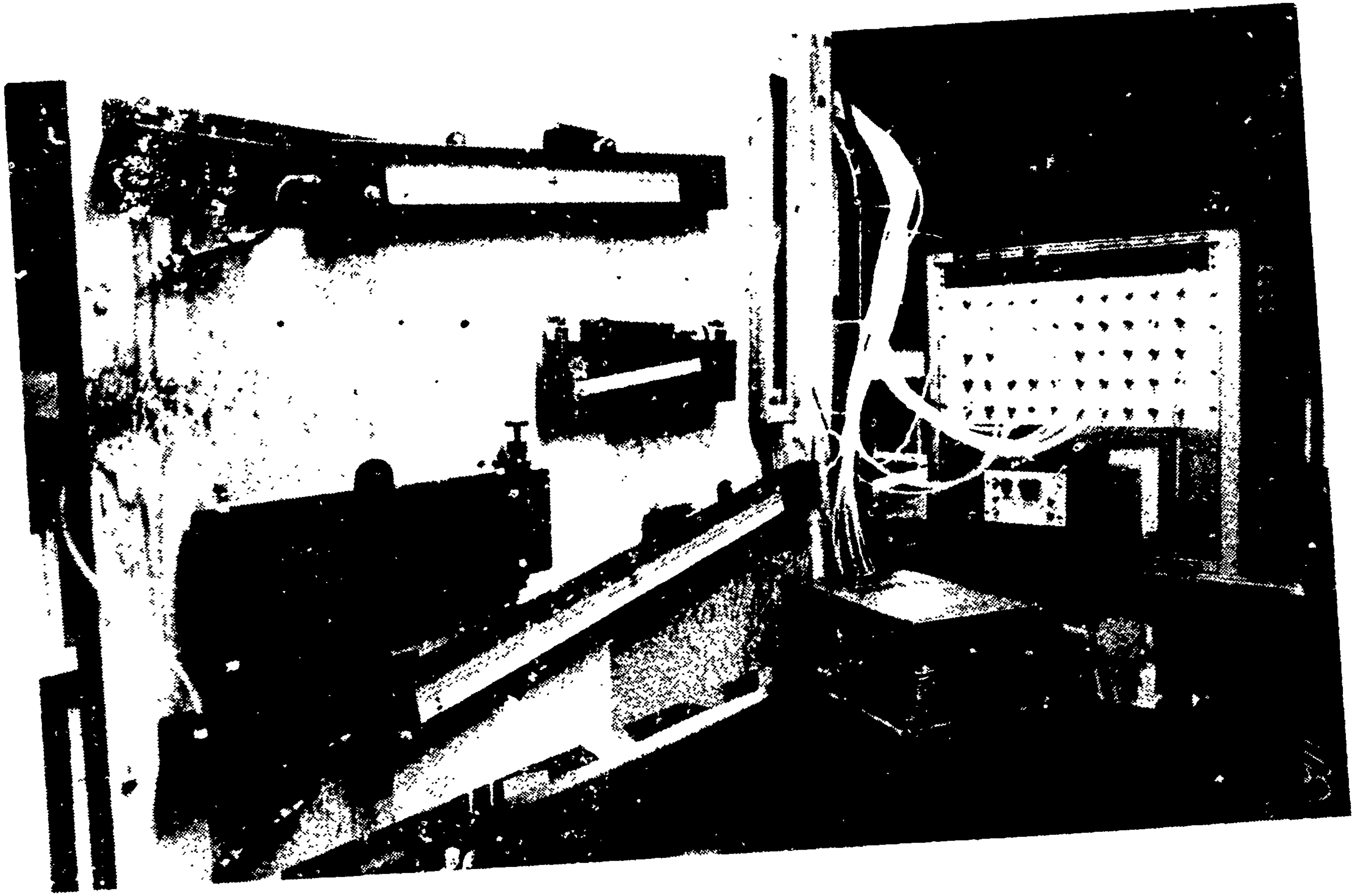

FIGURE 22 INSTRUMENTATION

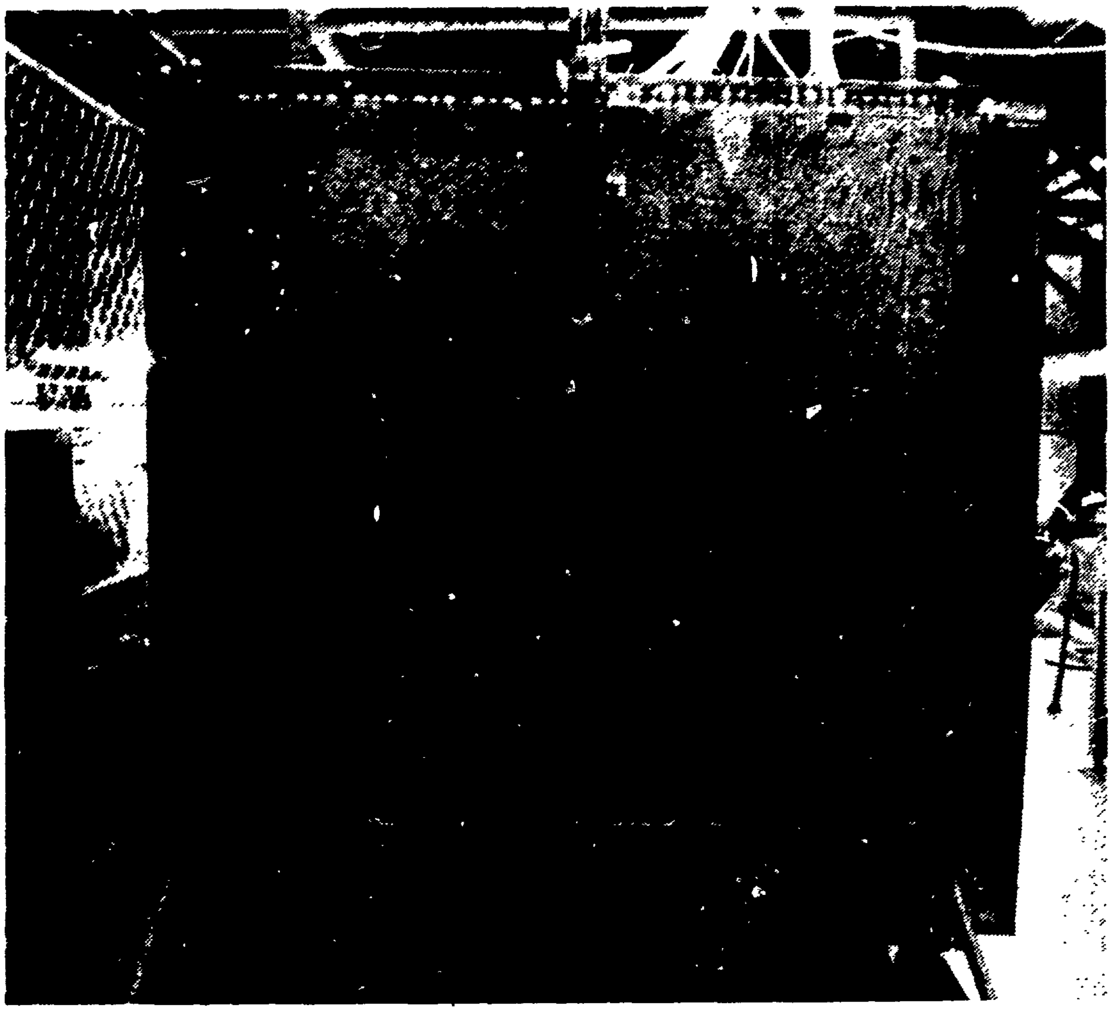

FIGURE 23. VELOCITY PROFILE APPARATUS

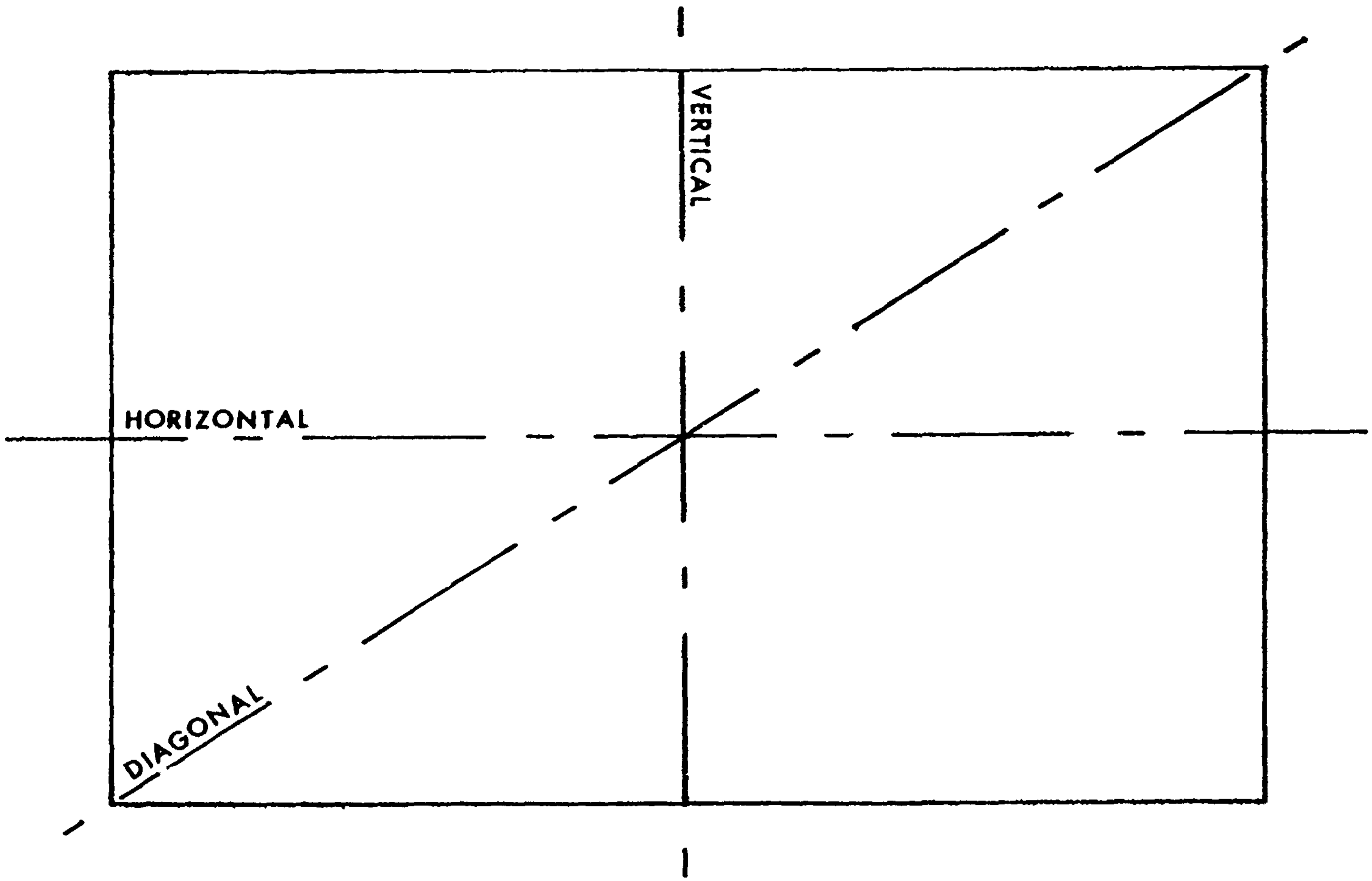

FIGURE 24. VELOCITY PROFILE TRAVERSE DIRECTIONS

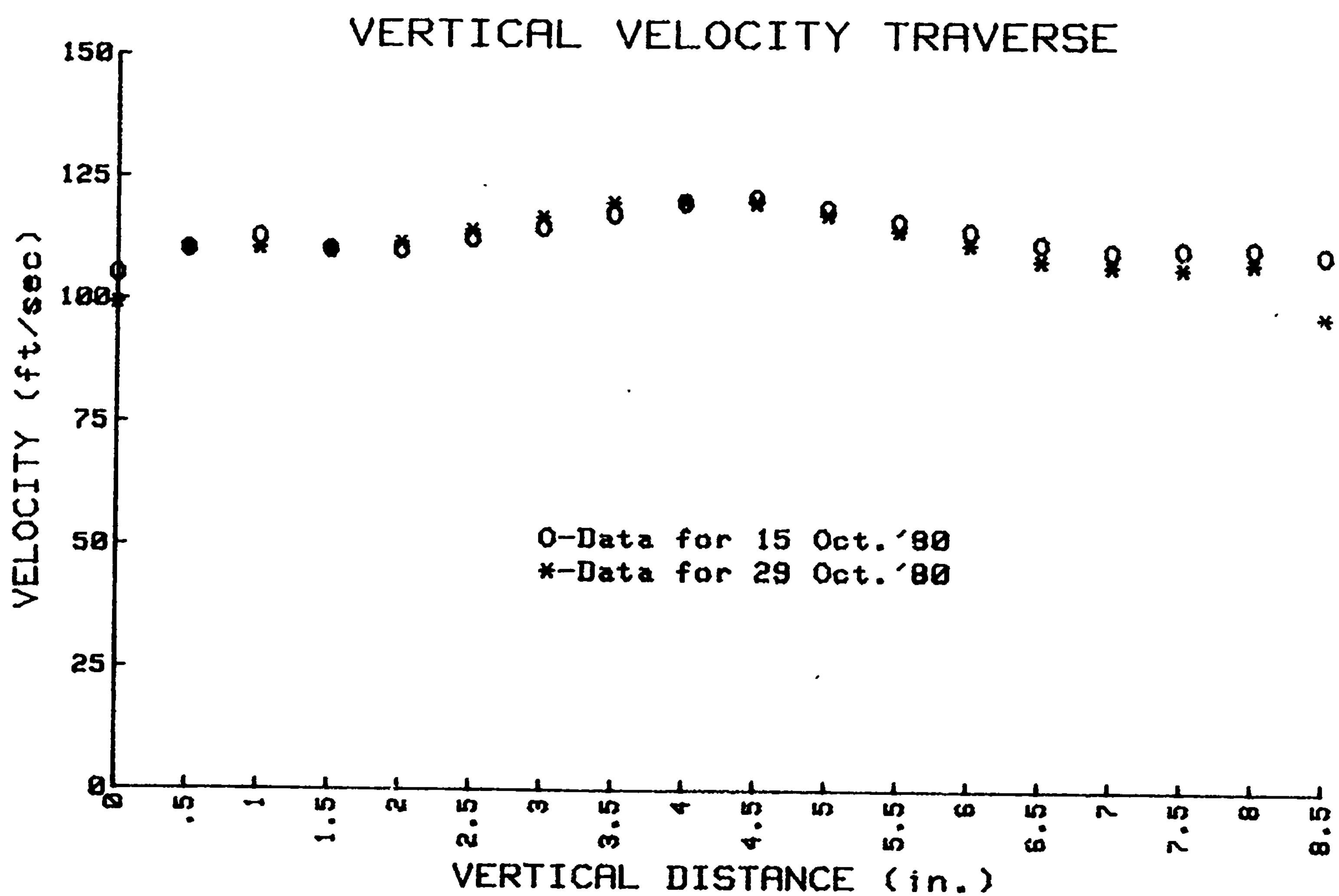

FIGURE 25a. EXIT VELOCITY PROFILE (VERTICAL TRAVERSE)

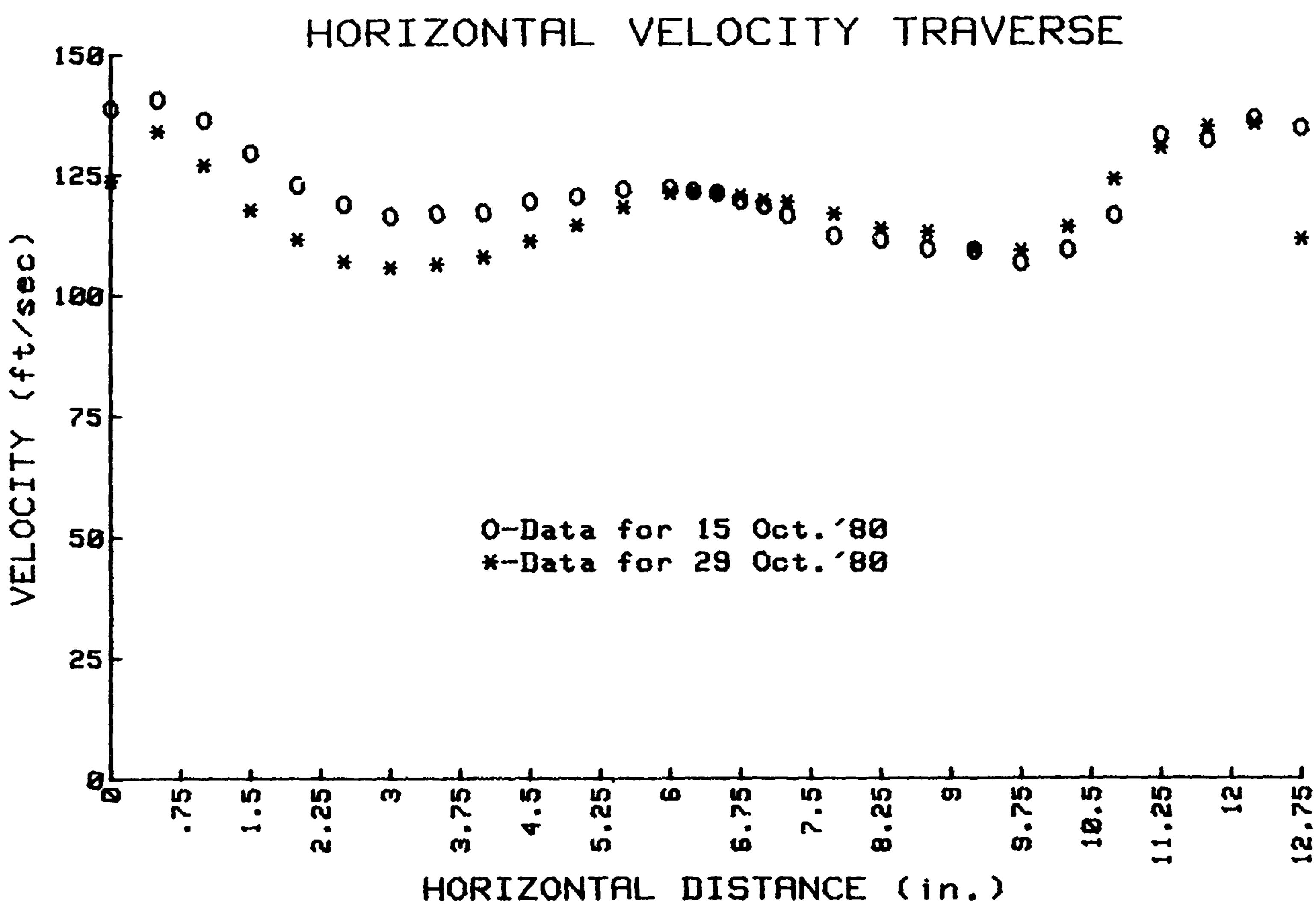

FIGURE 25b. EXIT VELOCITY PROFILE (HORIZONTAL TRAVERSE)

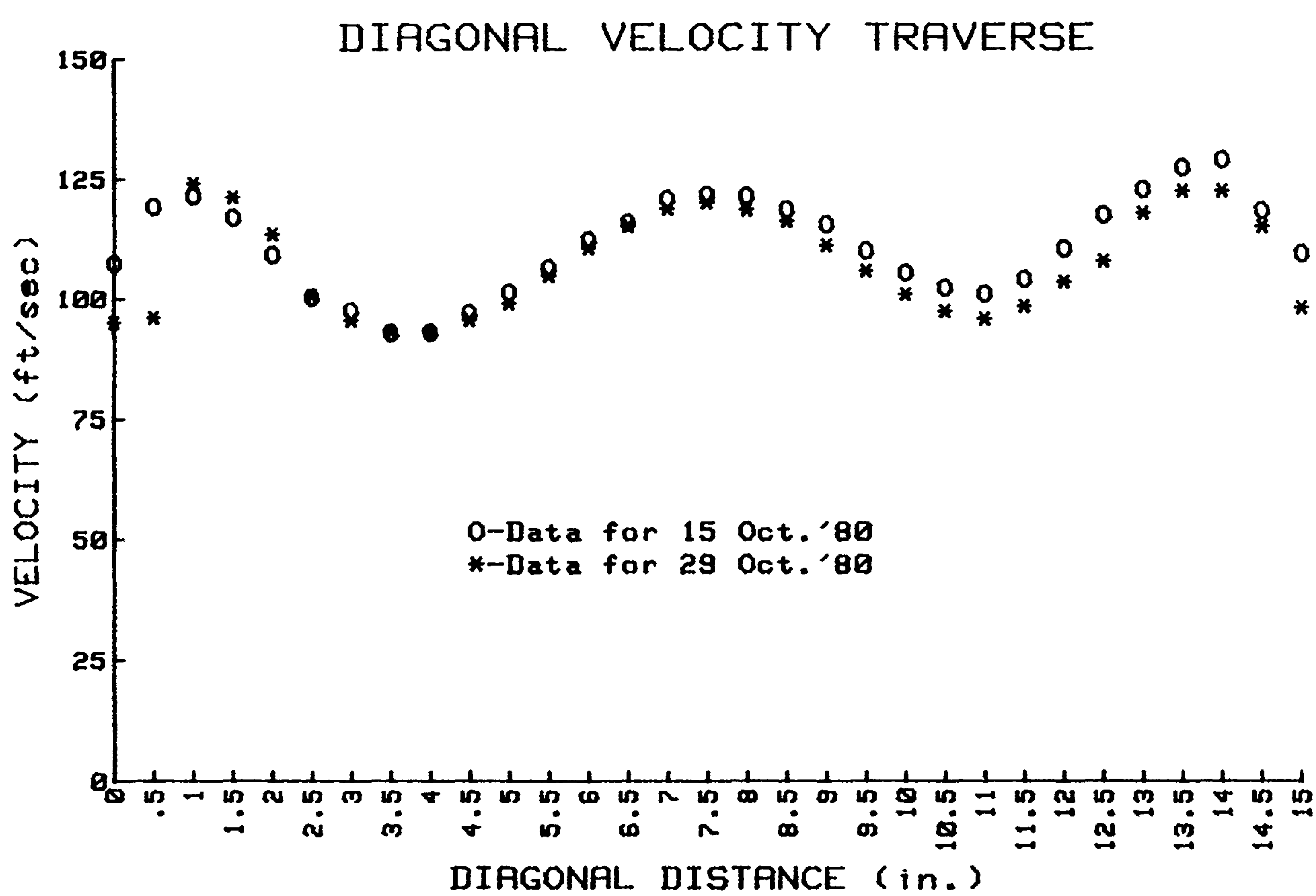

FIGURE 25c. EXIT VELOCITY PROFILE (DIAGONAL TRAVERSE)

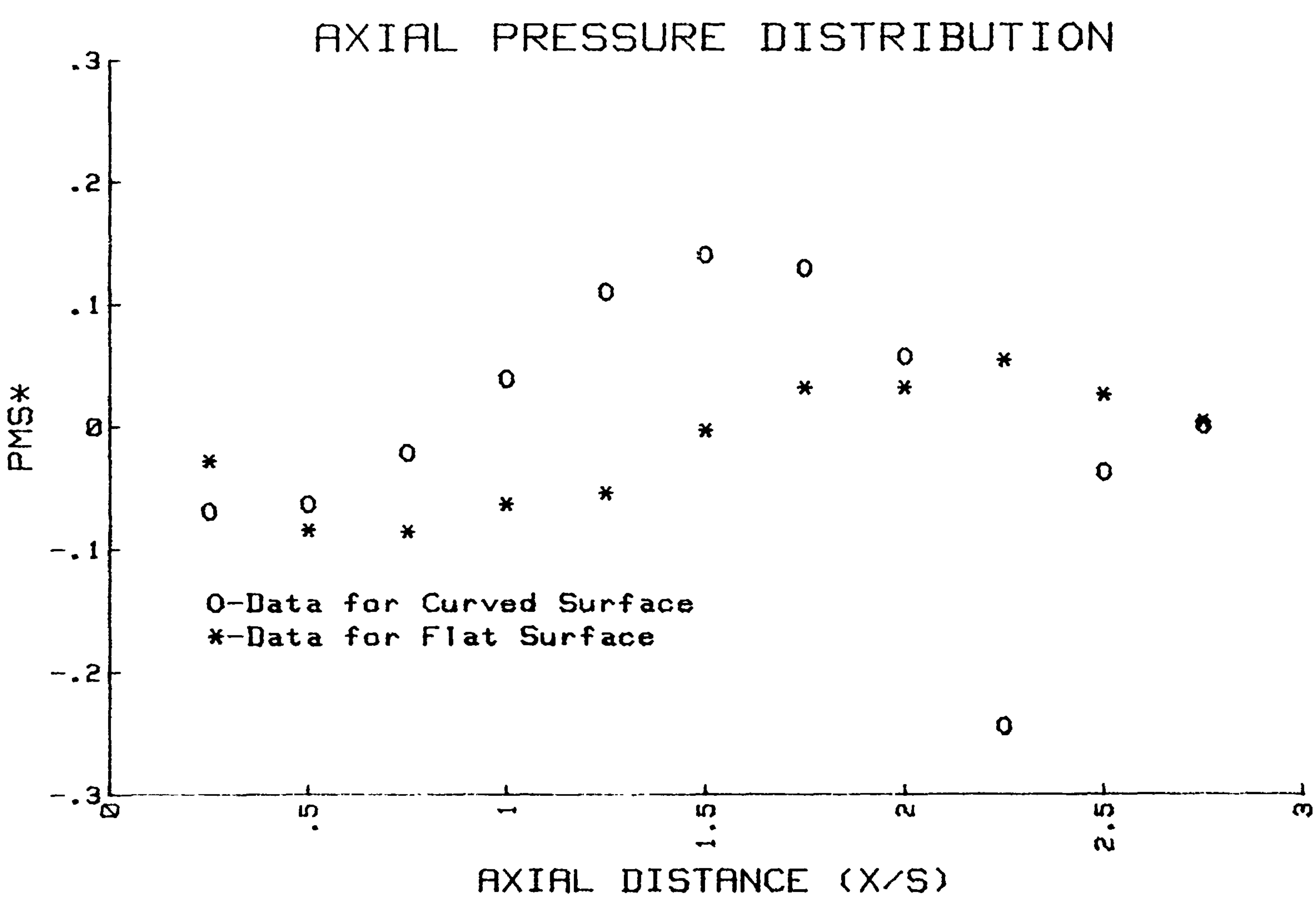

FIGURE 26a. MIXING STACK AXIAL PRESSURE DISTRIBUTION (RUN 1)

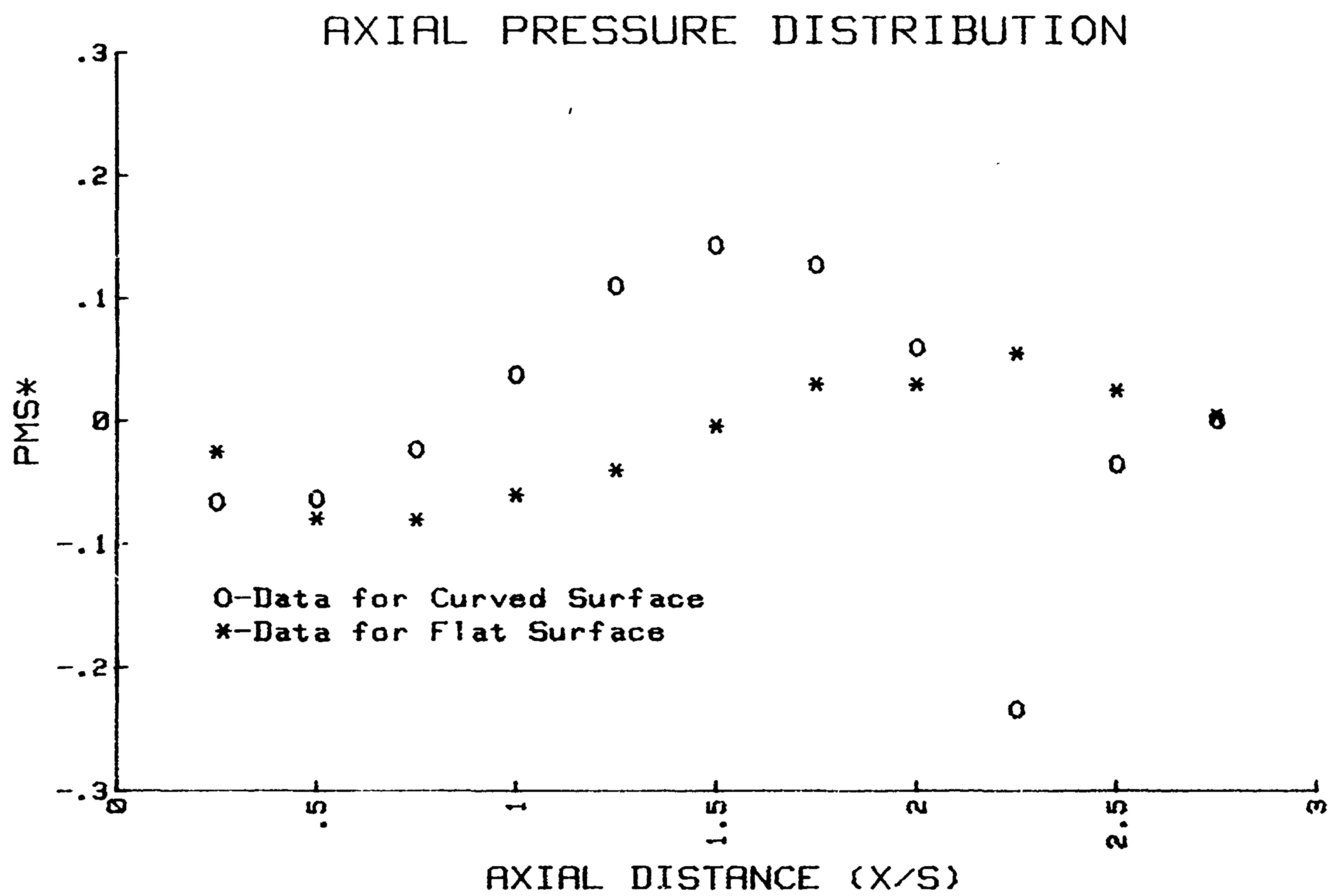

FIGURE 26b. MIXING STACK AXIAL PRESSURE DISTRIBUTION (RUN 2)

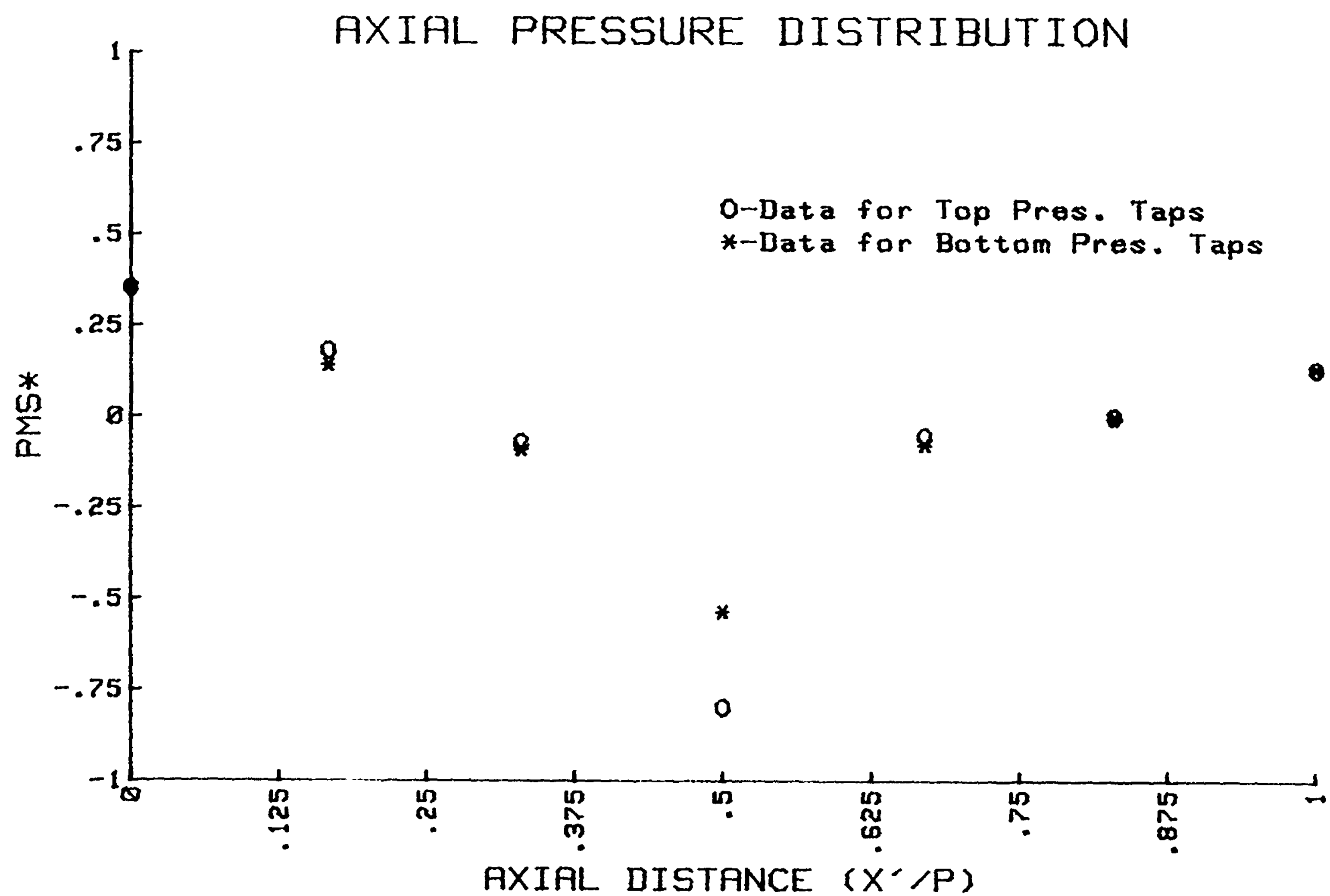

FIGURE 27a. SYMMETRIC PLUG AXIAL PRESSURE DISTRIBUTION (RUN 1)

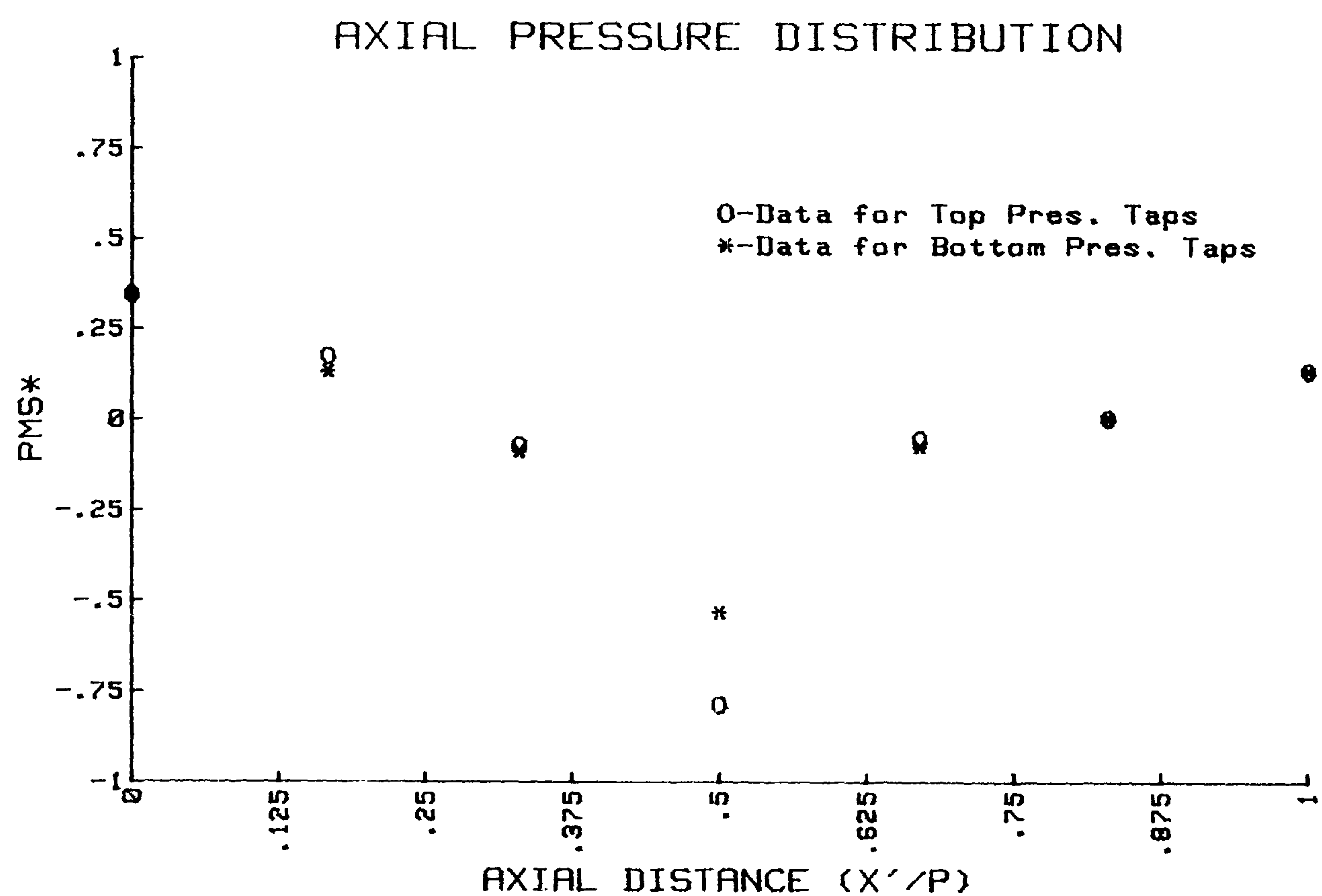

FIGURE 27b. SYMMETRIC PLUG AXIAL PRESSURE DISTRIBUTION (RUN 2)

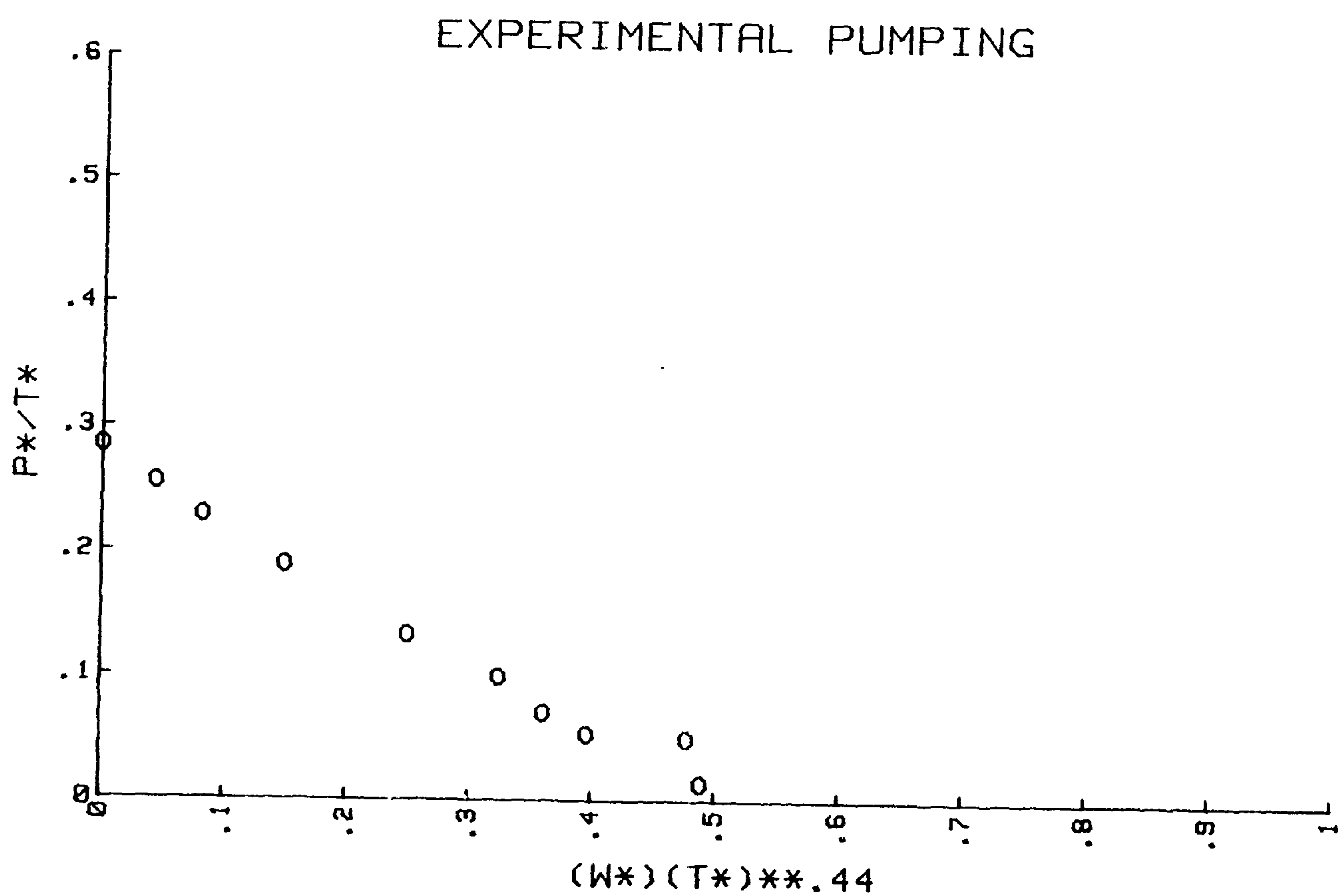

FIGURE 28a. ILLUSTRATIVE PLOT OF EXPERIMENTAL DATA CORRELATION IN EQUATION (14) (RUN 1)

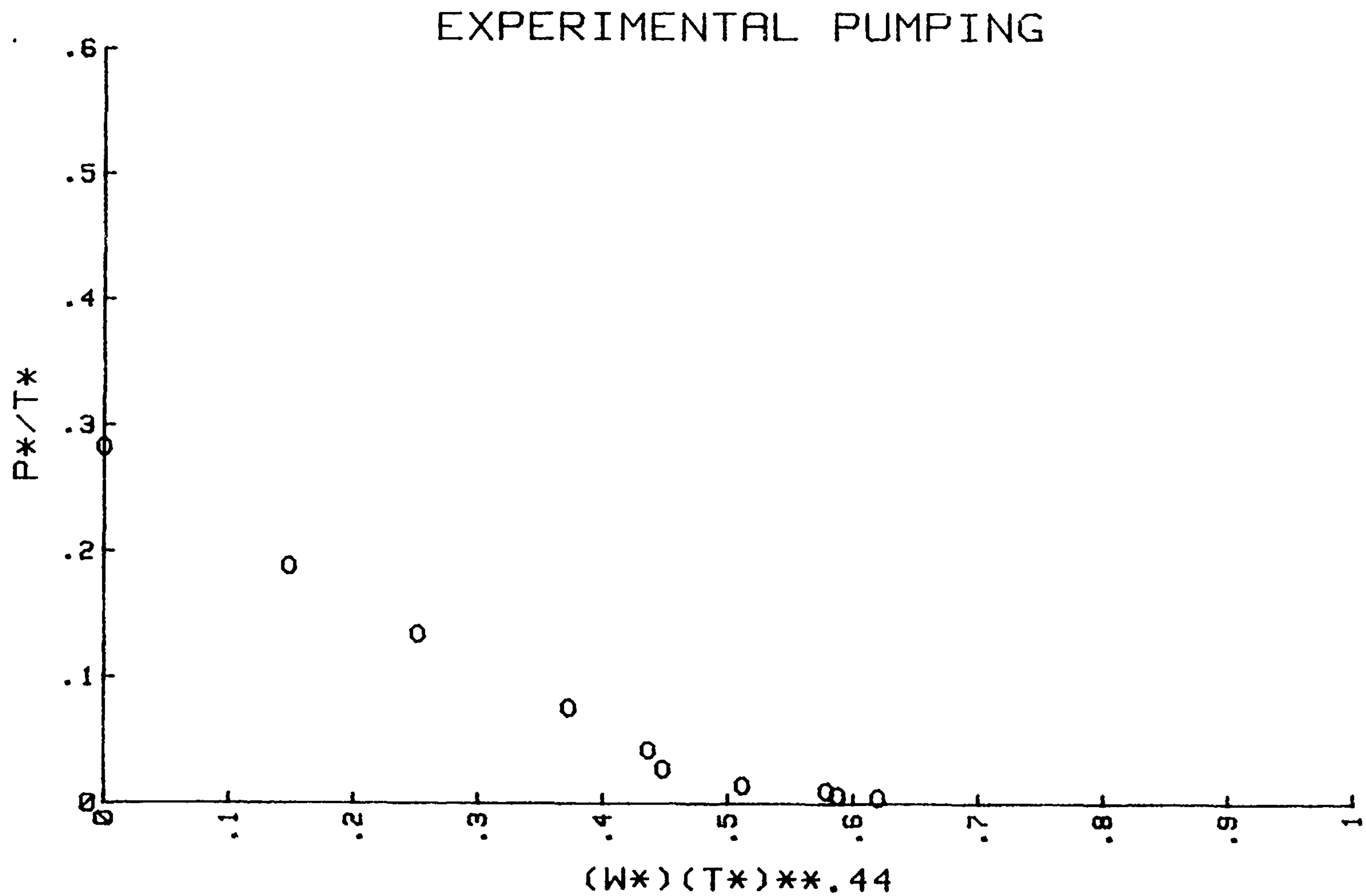

FIGURE 28b. ILLUSTRATIVE PLOT OF EXPERIMENTAL DATA CORRELATION IN EQUATION (14) (RUN 2)

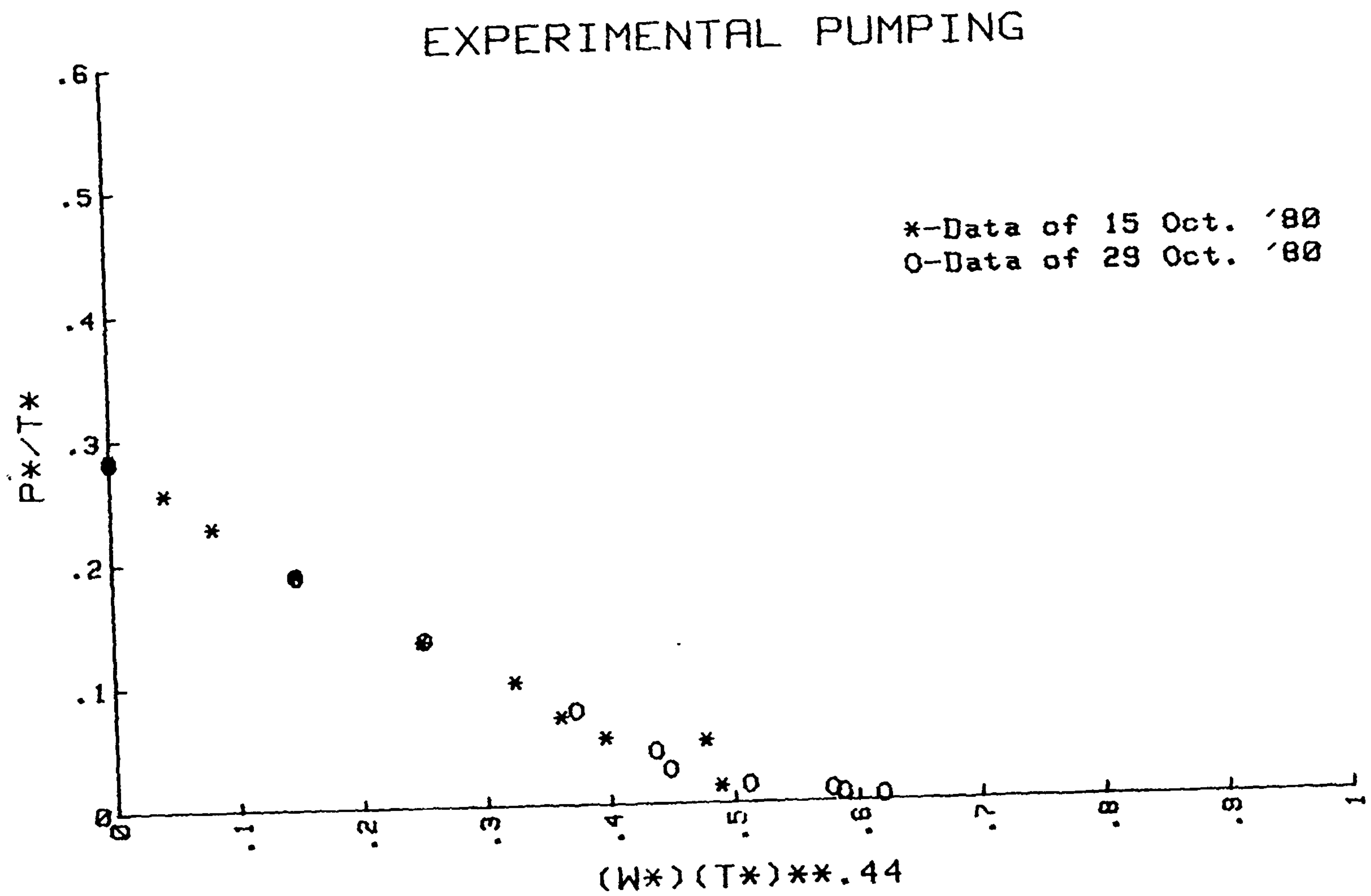

FIGURE 28c. ILLUSTRATIVE PLOT FOR COMPARISON OF EXPERIMENTAL DATA

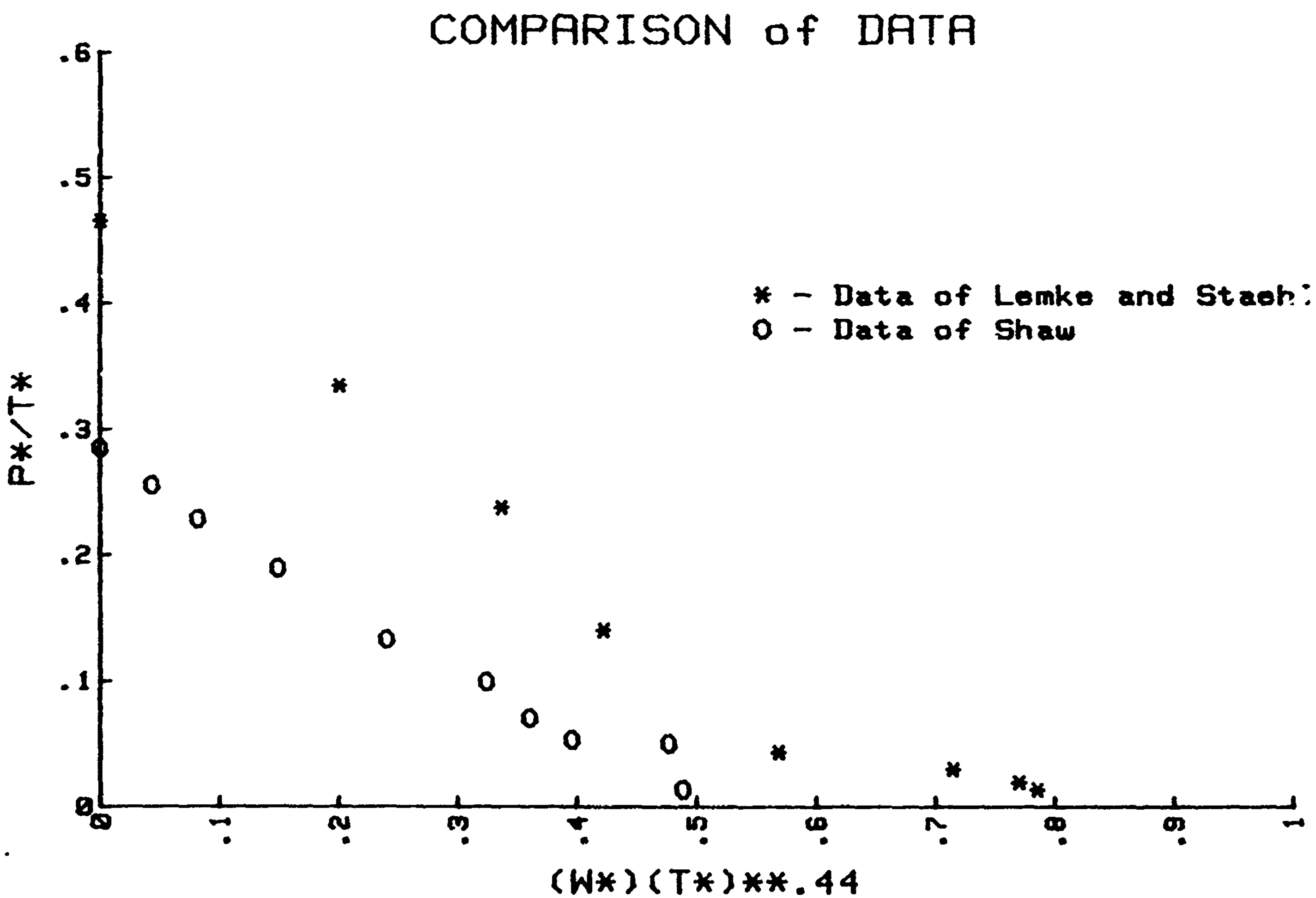

FIGURE 29. ILLUSTRATIVE PLOT FOR COMPARISON OF EXPERIMENTAL DATA

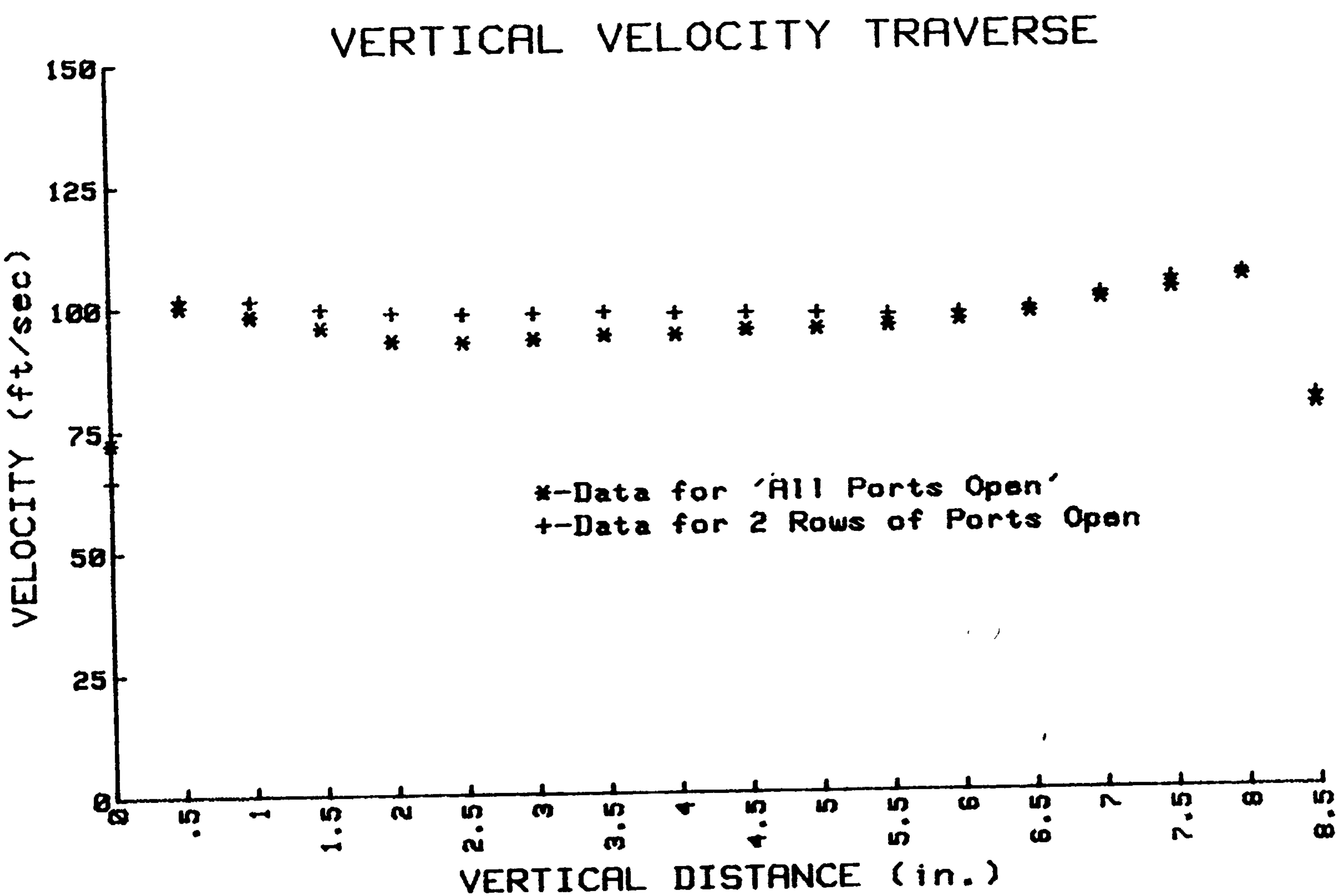

FIGURE 30a. EXIT VELOCITY PROFILE WITH TERTIARY AIR FLOW (VERTICAL TRAVERSE)

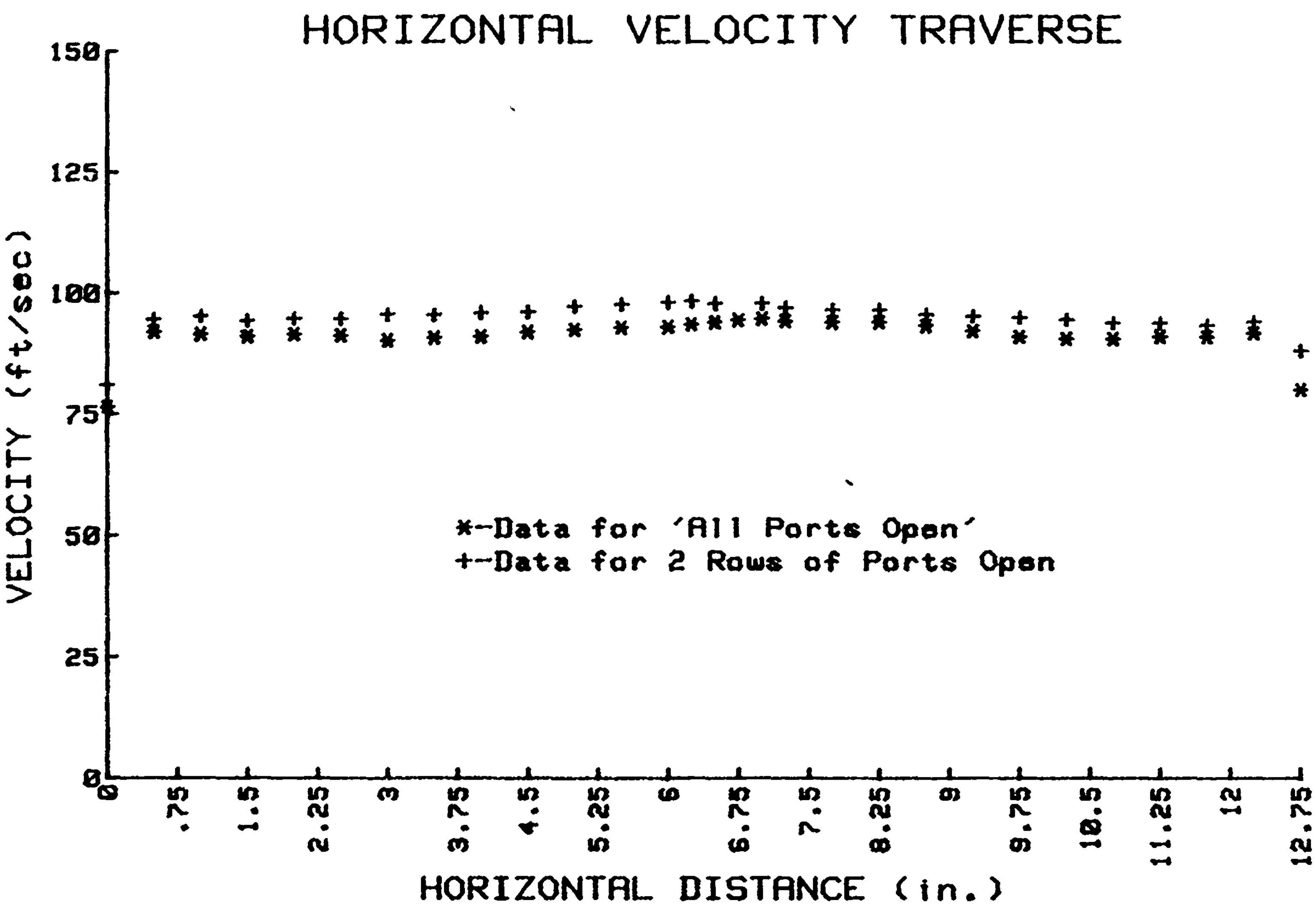

FIGURE 30b. EXIT VELOCITY PROFILE WITH TERTIARY AIR FLOW (HORIZONTAL TRAVERSE)

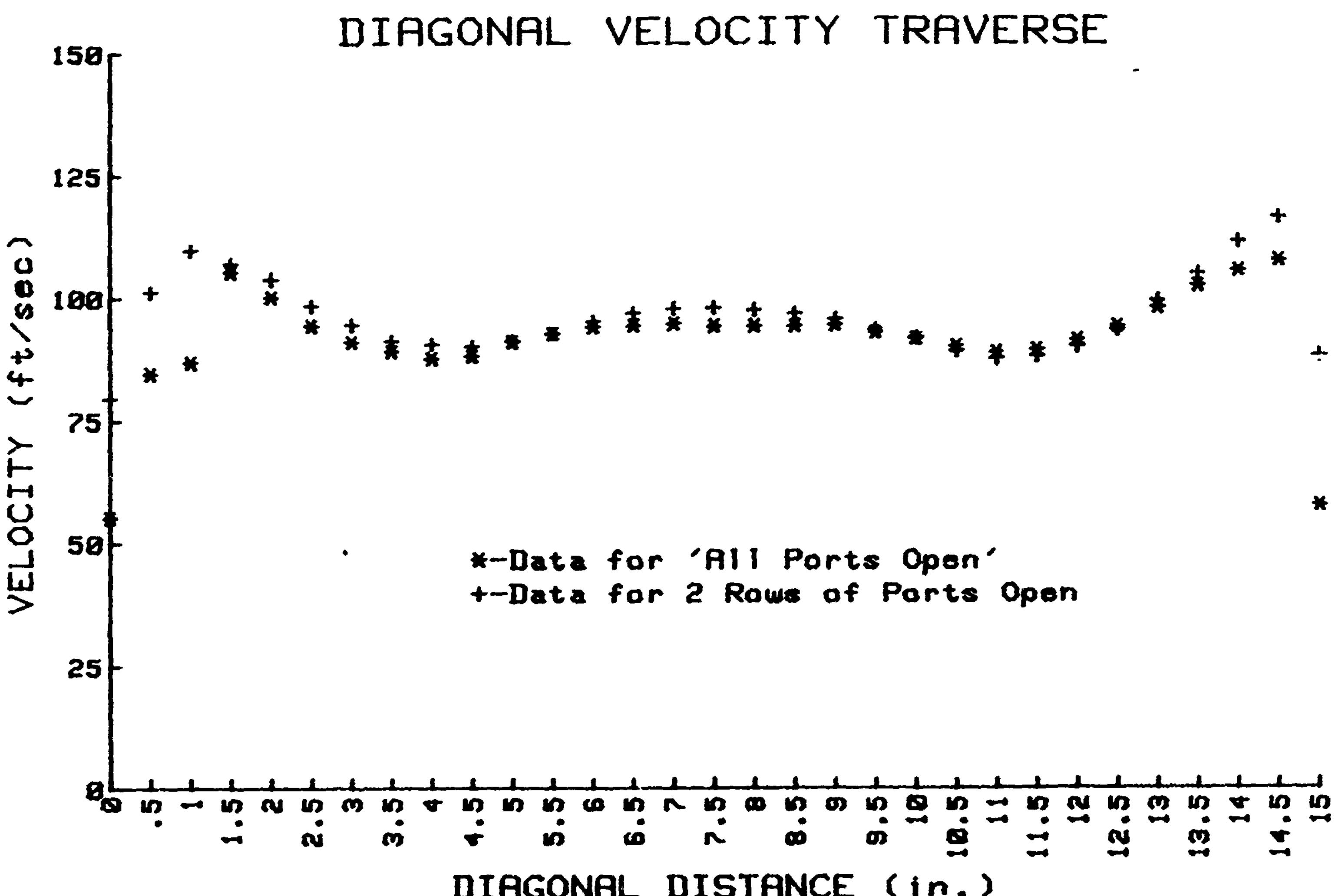

FIGURE 30c. EXIT VELOCITY PROFILE WITH TERTIARY AIR FLOW
(DIAGONAL TRAVERSE)

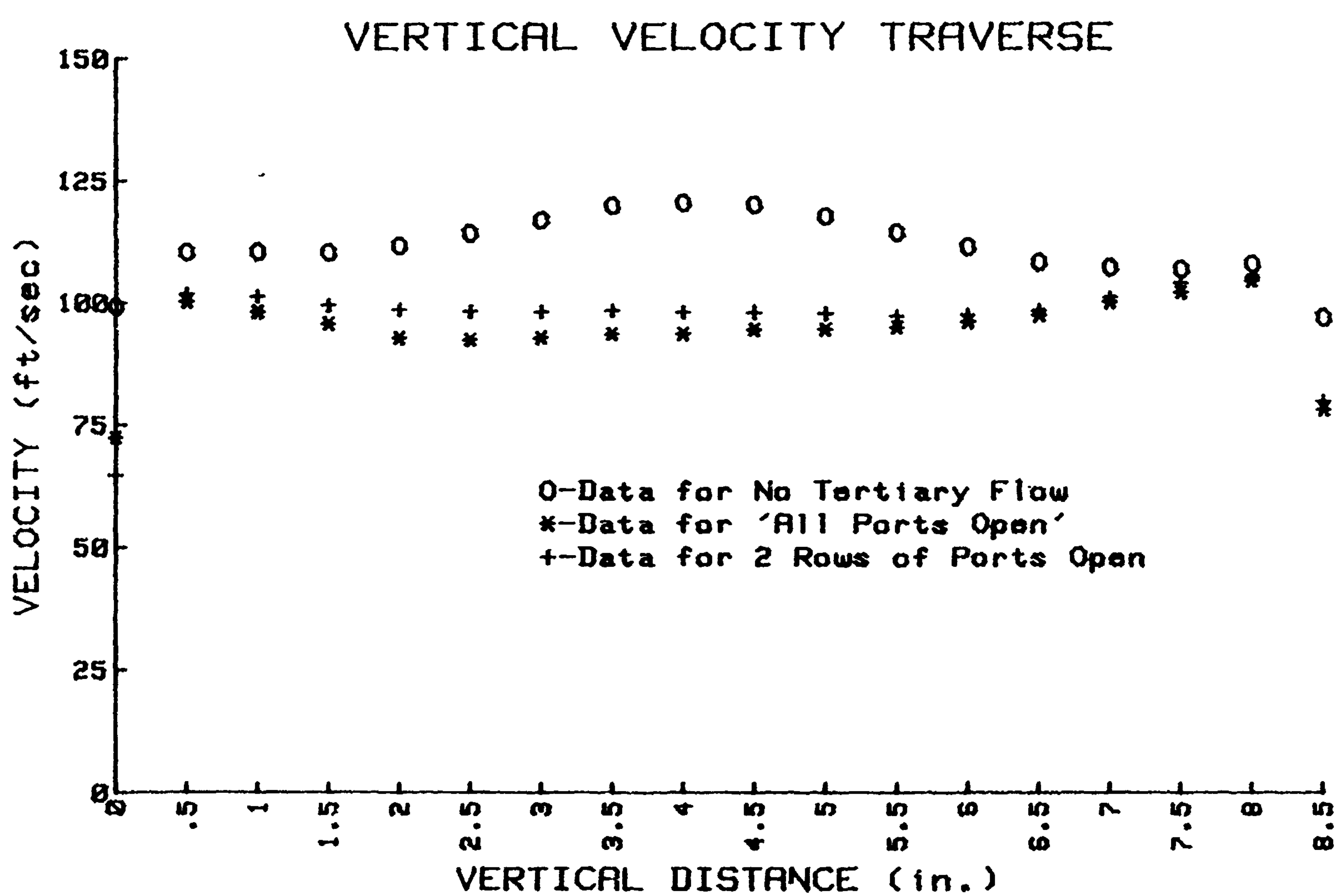

FIGURE 31a. ILLUSTRATIVE PLOT FOR COMPARISON OF EXIT VELOCITY PROFILE (VERTICAL TRAVERSE)

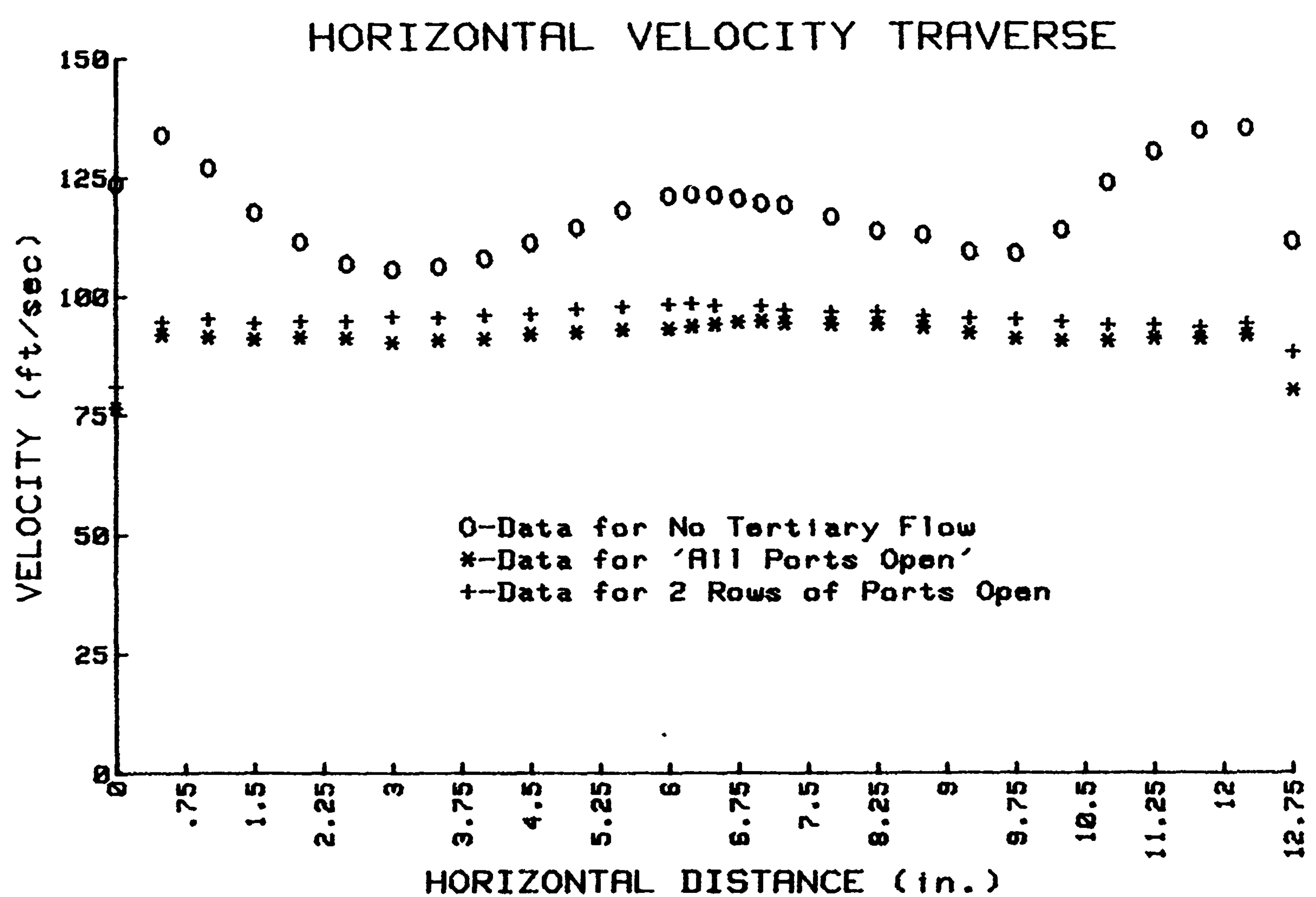

FIGURE 31b. ILLUSTRATIVE PLOT FOR COMPARISON OF EXIT VELOCITY PROFILE (HORIZONTAL TRAVERSE)

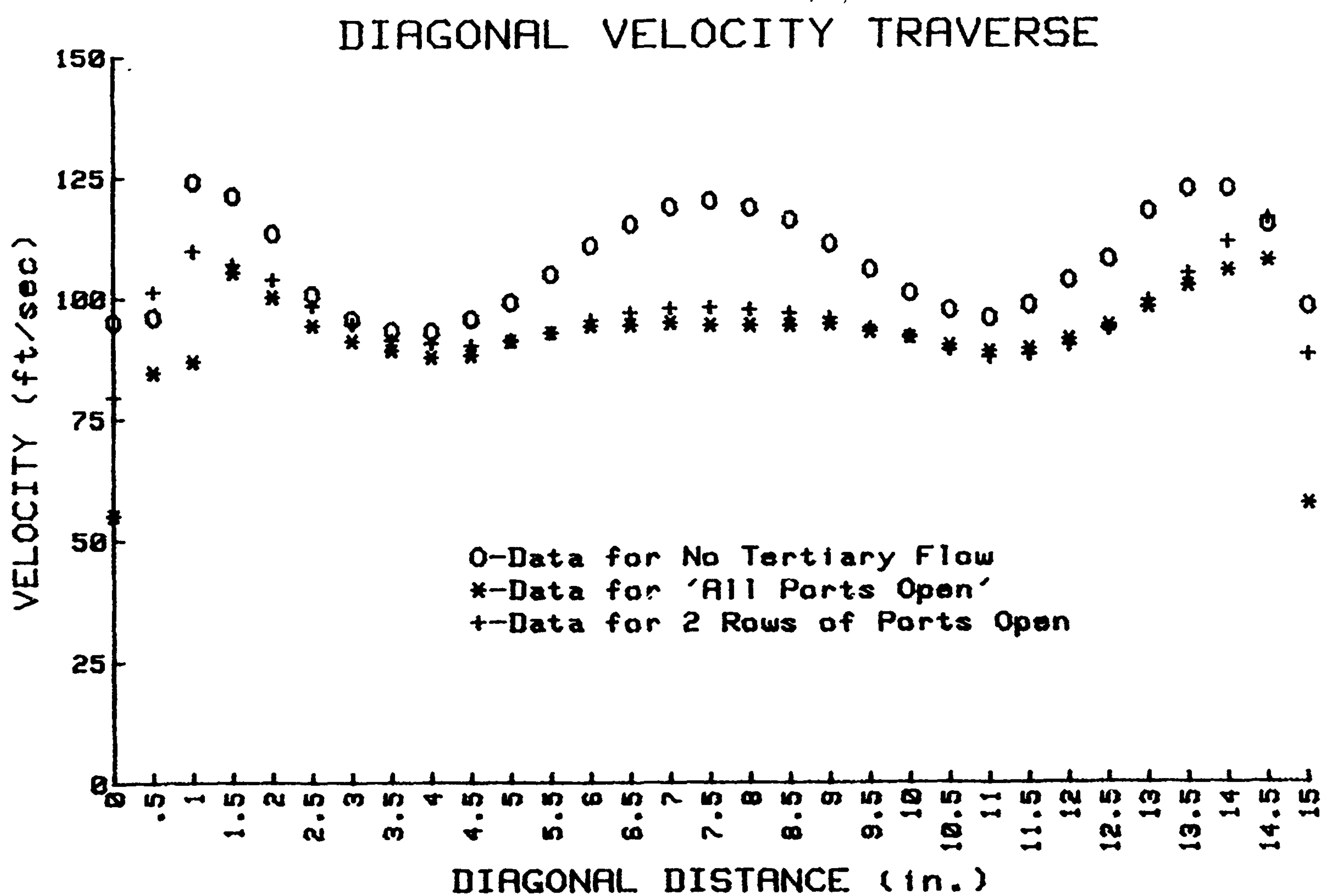

FIGURE 31c. ILLUSTRATIVE PLOT FOR COMPARISON OF EXIT VELOCITY PROFILE (DIAGONAL TRAVERSE)

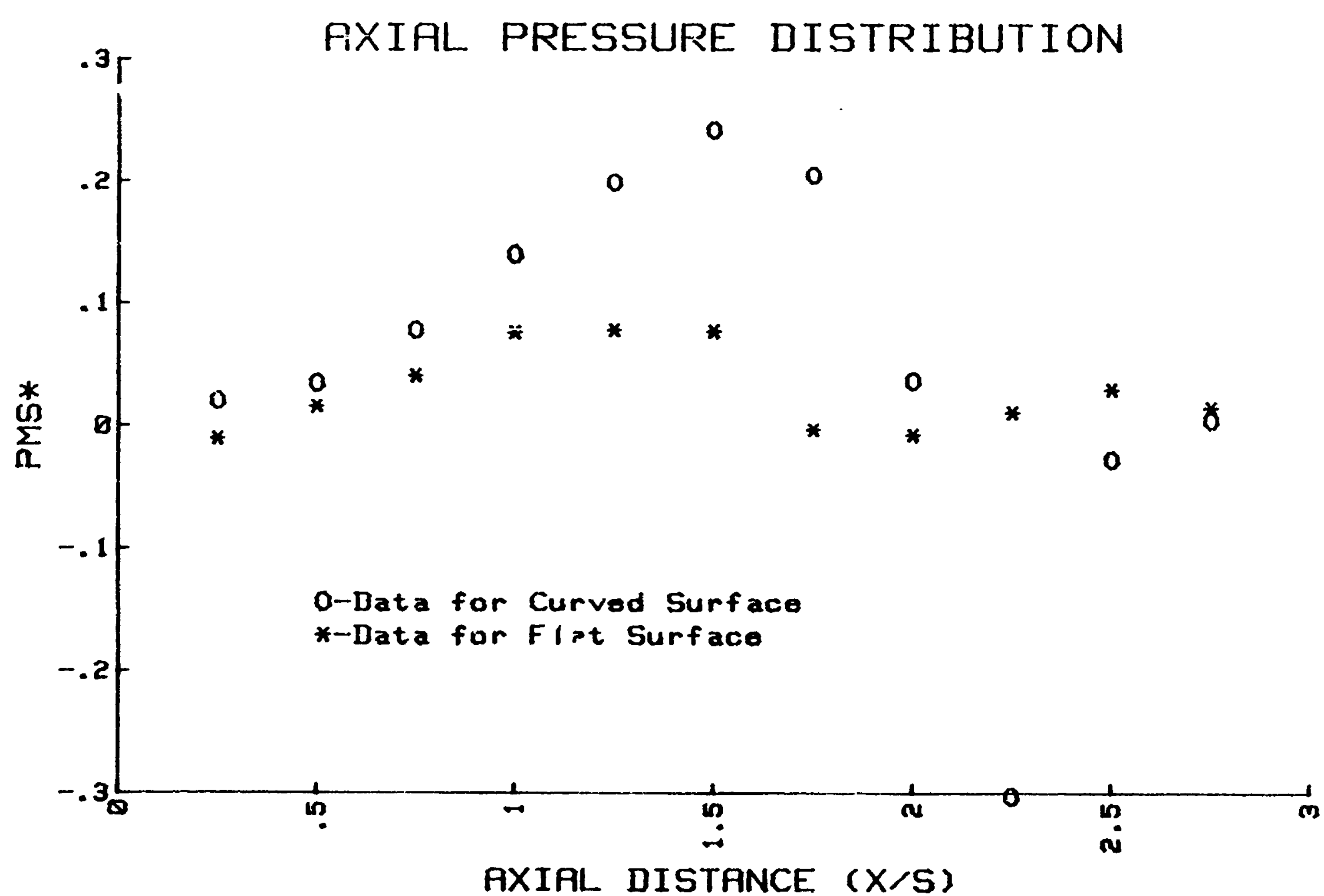

FIGURE 32a. MIXING STACK AXIAL PRESSURE DISTRIBUTION WITH TERTIARY AIR FLOW (ALL PORTS OPEN)

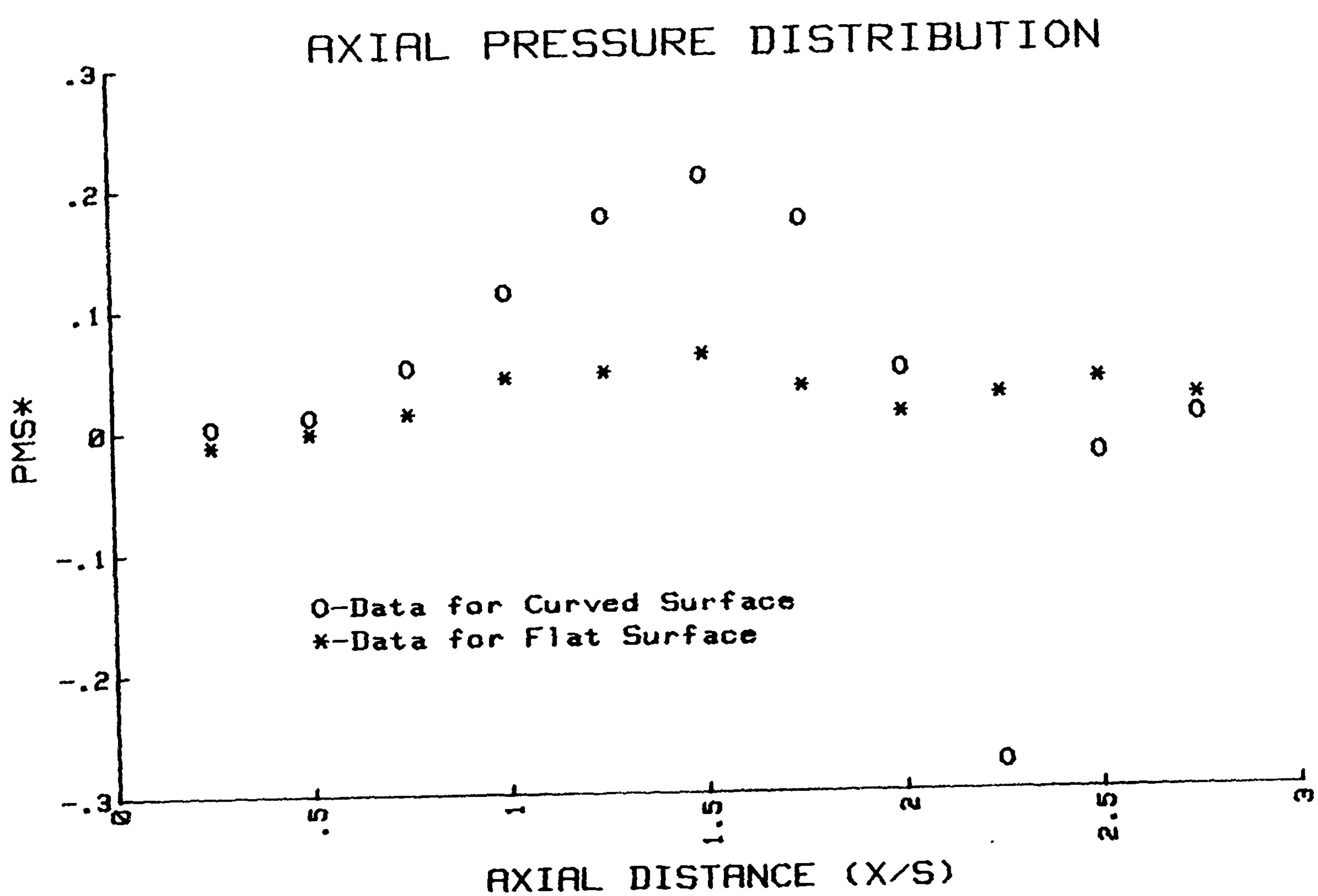

FIGURE 32b. MIXING STACK AXIAL PRESSURE DISTRIBUTION WITH TERTIARY AIR FLOW (2 ROWS OF PORTS OPEN)

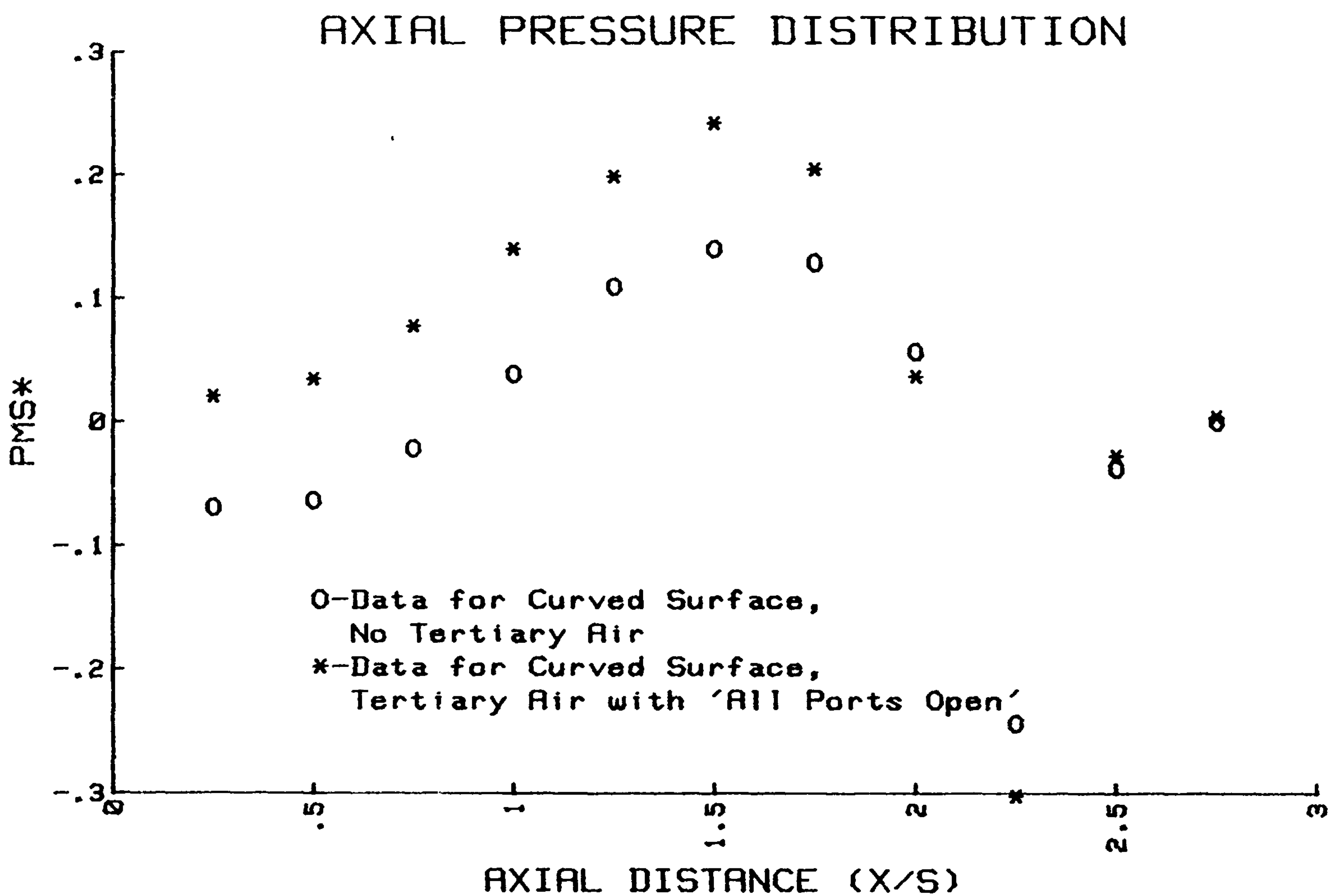

FIGURE 33a. ILLUSTRATIVE PLOT FOR COMPARISON OF AXIAL PRESSURE DISTRIBUTION

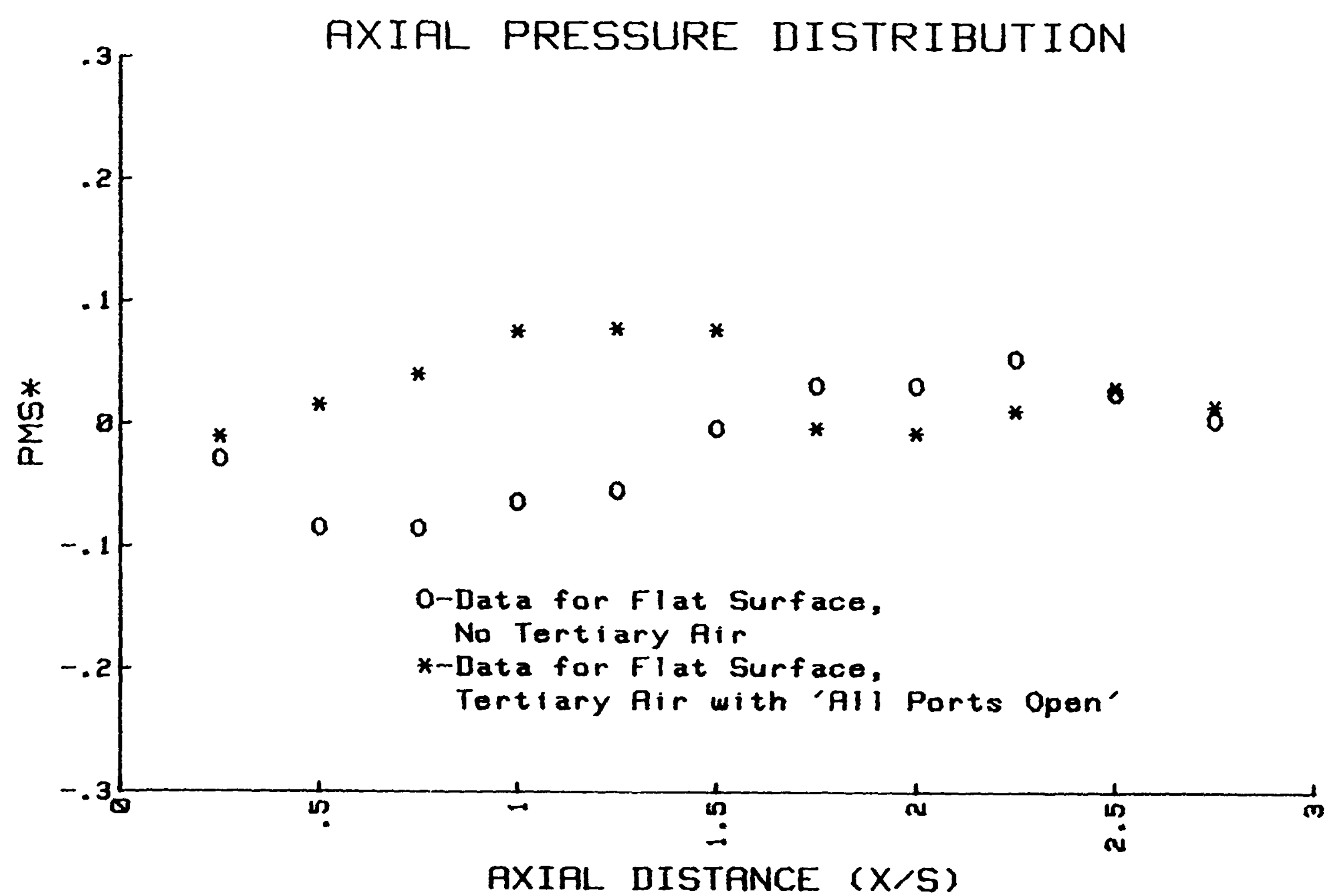

FIGURE 33b. ILLUSTRATIVE PLOT FOR COMPARISON OF AXIAL PRESSURE DISTRIBUTIONS

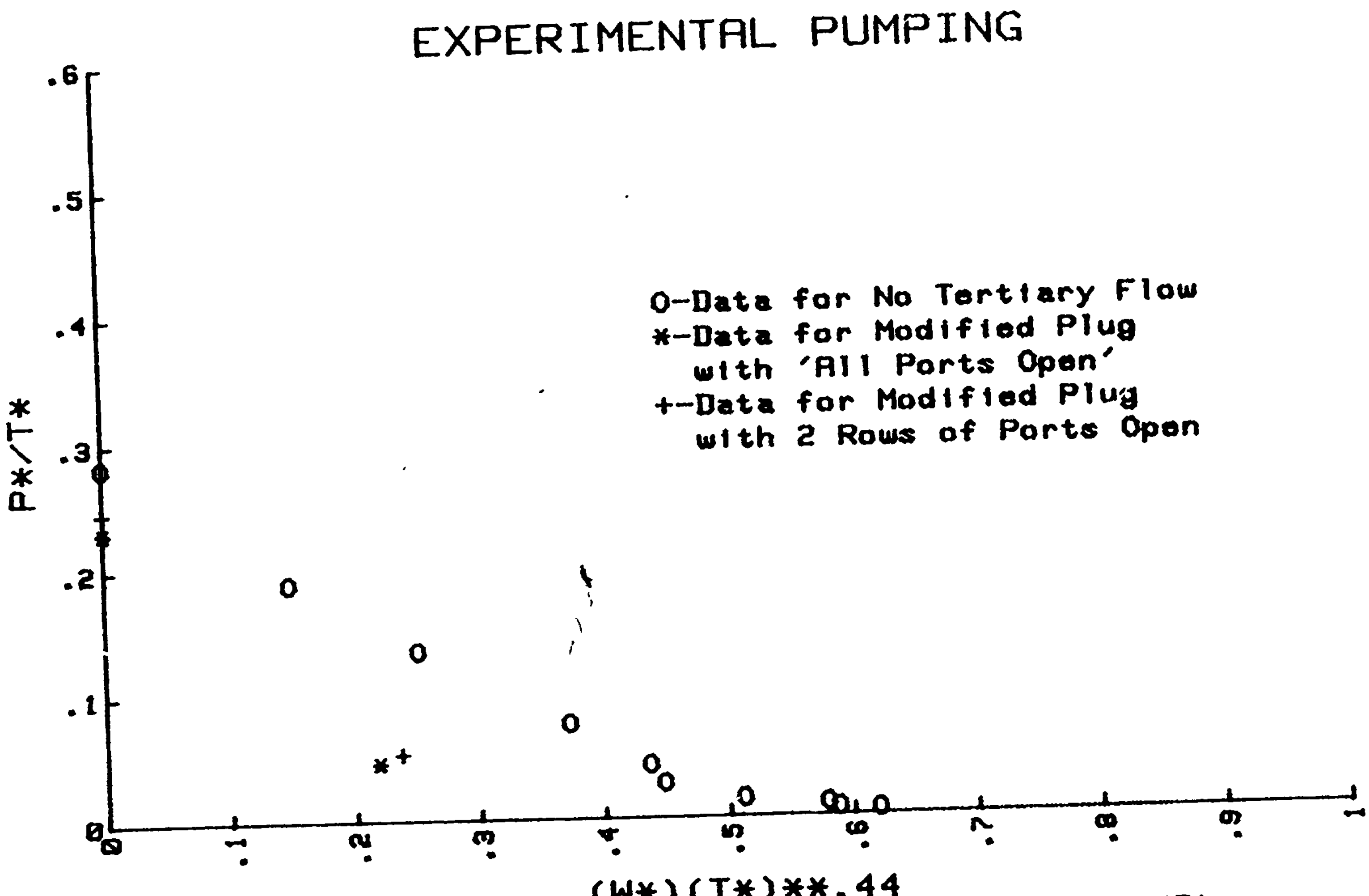

FIGURE 34. ILLUSTRATIVE PLOT FOR COMPARISON OF EXPERIMENTAL DATA

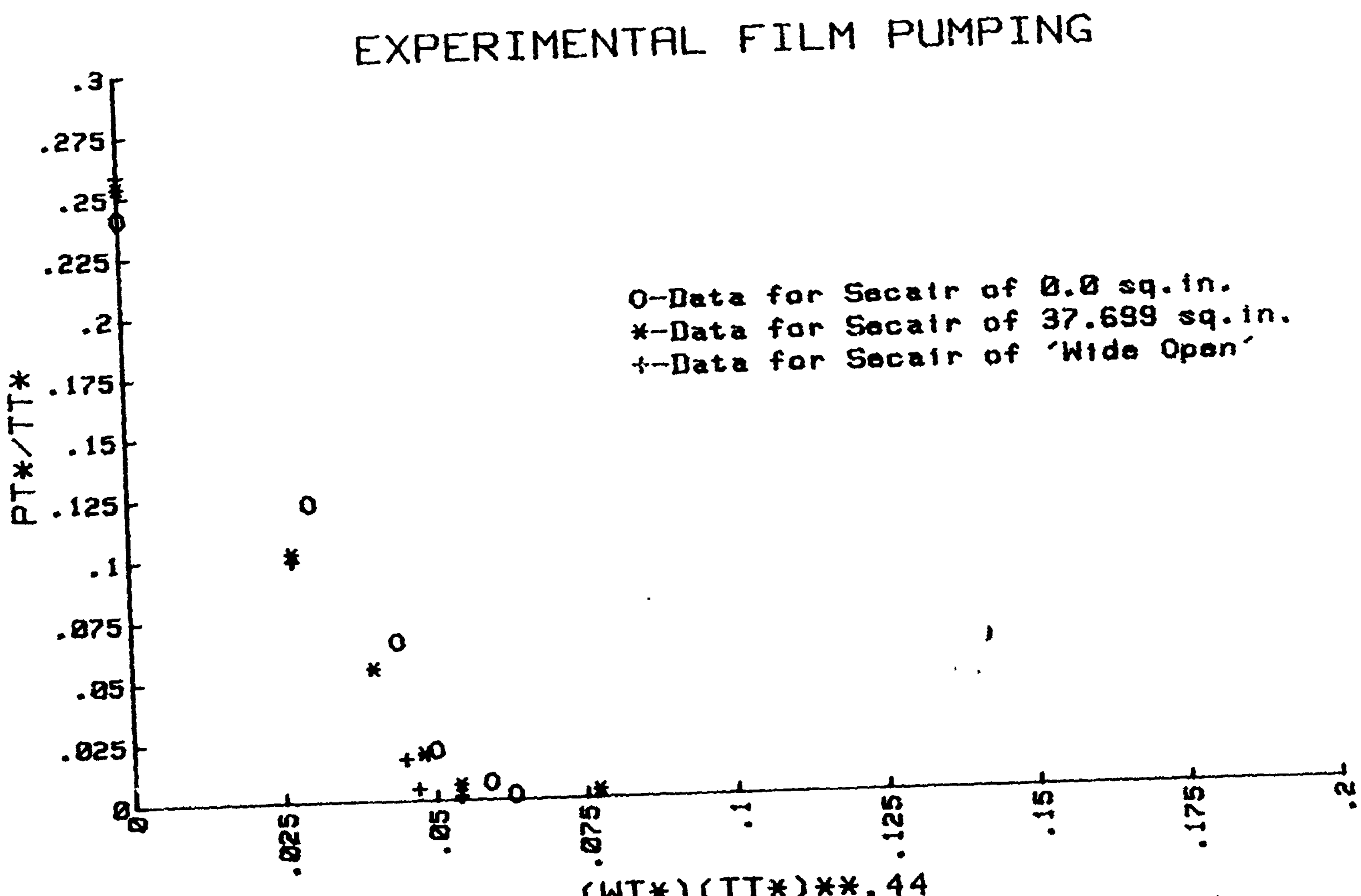

FIGURE 35a. ILLUSTRATIVE PLOT OF EXPERIMENTAL DATA CORRELATION IN EQUATION (14) FOR TERTIARY AIR FLOW (ALL PORTS OPEN)

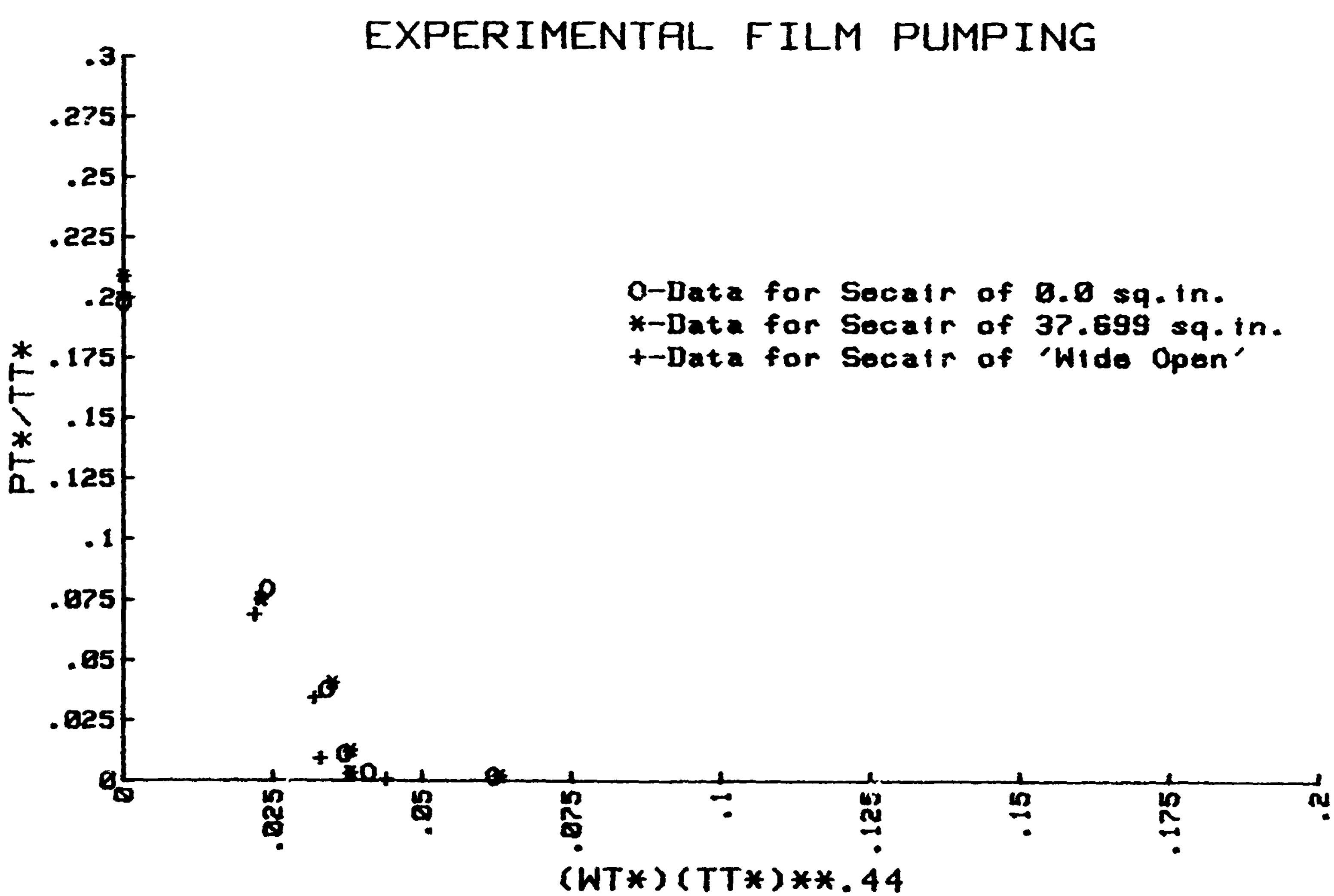

FIGURE 35b. ILLUSTRATIVE PLOT OF EXPERIMENTAL DATA CORRELATION IN EQUATION (14) FOR TERTIARY AIR FLOW (2 ROWS OF PORTS OPEN)

DATA TAKEN ON 15 OCTOBER 1980 BY R. SHAW
D/S = 0.5 : (PROOF RUN)
PAGE 1 OF 2

NUMBER OF PRIMARY NOZZLES: 4
PRIMARY NOZZLE AREA (AMP): 9.06 SQ. INCHES
MIXING STACK LENGH (L): 25.50 INCHES
MIXING STACK DIMENSION (S): 8.50 INCHES
MIXING STACK L/S: 3.00

UPTAKE AREA (AU): 108.375 SQ. INCHES
AREA RATIO, AM/AP: 2.99
ORIFICE DIAMETER: 6.902 INCHES
ORIFICE BETA: 0.497
AMBIENT PRESSURE: 30.06 INCHES HG

N RUN	POR (INCHES OF WATER)	DPOR (INCHES OF WATER)	TOR (DEGREES FAHRENHEIT)	TUPT (DEGREES FAHRENHEIT)	TAMB (DEGREES FAHRENHEIT)	PU-PA (INCHES OF WATER)	PA-PS (INCHES OF WATER)	PA-PNZ (INCHES OF WATER)	SECONDARY AREA SQUARE INCHES
1	.7	21.8	57.3	110.8	82.5	7.15	2.72	2.72	0.0
2	0.7	22.1	59.9	114.8	82.5	7.50	2.48	2.48	3.142
3	0.7	22.2	61.3	114.7	82.5	7.75	2.22	2.21	6.283
4	0.7	22.0	60.8	114.9	82.5	8.10	1.82	1.82	12.566
5	0.7	22.1	60.7	114.4	82.5	8.60	1.28	1.28	25.133
6	0.7	22.0	60.5	115.1	82.5	8.90	0.95	0.96	37.699
7	0.7	21.9	62.5	115.9	82.5	9.05	0.67	0.66	50.265
8	0.7	21.7	62.9	116.2	82.5	9.20	0.50	0.50	63.460
9	0.7	21.7	62.4	116.6	82.5	9.25	0.47	0.47	78.540
10	0.7	22.0	61.2	115.3	82.5	9.70	0.13	0.13	157.080
11	0.7	22.1	61.3	114.4	82.5	9.80	0.00	0.00	**********

N RUN	W*	P*	T*	P*/T*	W*T**.44	WP LBM/SEC	WS LBM/SEC	UP FT/SEC	UM FT/SEC	UU FT/SEC	UPT MACH
1	0.0	0.2709	0.9534	0.2850	0.0	3.736	0.0	213.81	71.53	71.53	0.061
2	0.0445	0.2413	0.9438	0.2557	0.0434	3.753	0.167	216.10	75.32	72.30	0.062
3	0.0840	0.2159	0.9439	0.2287	0.0819	3.756	0.315	216.12	78.01	72.31	0.062
4	0.1528	0.1791	0.9436	0.1898	0.1489	3.741	0.572	215.12	82.30	71.97	0.061
5	0.2555	0.1259	0.9444	0.1333	0.2492	3.750	0.958	215.15	89.30	71.98	0.061
6	0.3324	0.0937	0.9433	0.0994	0.3239	3.742	1.244	214.80	94.34	71.86	0.061
7	0.3700	0.0666	0.9420	0.0707	0.3604	3.726	1.379	214.05	96.53	71.61	0.061
8	0.4065	0.0502	0.9415	0.0533	0.3959	3.708	1.507	213.01	98.51	71.27	0.061
9	0.4900	0.0471	0.9408	0.0500	0.4770	3.710	1.818	213.25	104.20	71.34	0.061
10	0.5014	0.0129	0.9430	0.0137	0.4886	3.739	1.875	214.30	105.58	71.70	0.061
11	********	0.0005	0.9444	0.0005	********	3.747	*******	214.36	******	71.72	0.061

TABLE Ia PERFORMANCE DATA OF MIXING STACK (BASE DESIGN, RUN NO. 1)

DATA TAKEN ON 15 OCTOBER 1980 BY R. SHAW
D/S = 0.5 : (PROOF RUN)
PAGE 2 OF 2

MIXING STACK PRESSURE DISTRIBUTION FOR RUN: 11 POSITION T

X/S:	0.25	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75
PMS(IN. H2O):	-0.695	-0.635	-0.215	0.390	1.115	1.410	1.305	0.580	-2.460	-0.375	0.010
PMS*:	-0.069	-0.063	-0.021	0.039	0.110	0.140	0.129	0.057	-0.244	-0.037	0.001

PLUG PRESSURE DISTRIBUTION FOR RUN: 11 POSITION T

X/P:	0.0	0.167	0.330	0.500	0.670	0.830	1.000
PLUG (IN. H2O):	3.560	1.840	-0.695	-8.050	-0.525	0.030	1.330
PP*:	0.353	0.182	-0.069	-0.798	-0.052	0.003	0.132

MIXING STACK PRESSURE DISTRIBUTION FOR RUN: 11 POSITION S

X/S:	0.25	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75
PMS(IN. H2C):	-0.280	-0.850	-0.860	-0.635	-0.540	-0.030	0.320	0.320	0.540	0.265	0.045
PMS*:	-0.028	-0.084	-0.085	-0.063	-0.054	-0.003	0.032	0.032	0.054	0.026	0.004

PLUG PRESSURE DISTRIBUTION FOR RUN: 11 POSITION B

X/P:	0.0	0.167	0.330	0.500	0.670	0.830	1.000
PLUG (IN. H2O):	3.560	1.440	-0.890	-5.400	-0.780	-0.020	1.330
PP*:	0.353	0.143	-0.088	-0.535	-0.077	-0.002	0.132

TABLE Ib AXIAL PRESSURE DISTRIBUTION DATA (BASE DESIGN, RUN NO. 1)

DATA TAKEN ON 29 OCTOBER 1980 BY R. SHAW
D/S = 0.5
PAGE 1 OF 2

NUMBER OF PRIMARY NOZZLES: 4
PRIMARY NOZZLE AREA (ANP): 9.05 SQ. INCHES
MIXING STACK LENGH (L): 25.50 INCHES
MIXING STACK DIMENSION (S): 8.50 INCHES
MIXING STACK L/S: 3.00

UPTAKE AREA (AU): 108.375 SQ. INCHES
AREA RATIO, AM/AP: 2.99
ORIFICE DIAMETER: 6.902 INCHES
ORIFICE BETA: 0.497
AMBIENT PRESSURE: 30.38 INCHES HG

N RUN	POR	DPOR	TOR	TUPT	TAMB	PU-PA	PA-PS	PA-PNZ	SECONDARY AREA
	INCHES OF WATER		DEGREES FAHRENHEIT			INCHES OF WATER			SQUARE INCHES
1	0.7	22.1	64.9	112.2	80.0	7.20	2.70	2.71	0.0
2	0.7	21.9	66.2	115.9	80.0	8.05	1.79	1.78	12.566
3	0.7	22.1	67.6	118.9	80.0	8.65	1.29	1.29	25.133
4	0.7	22.0	68.3	120.3	80.0	9.10	0.72	0.71	50.265
5	0.7	21.9	68.3	121.0	80.0	9.40	0.40	0.40	78.540
6	0.7	22.0	68.7	121.3	80.0	9.50	0.26	0.26	100.531
7	0.7	21.8	69.0	121.7	80.0	9.60	0.14	0.14	157.080
8	0.7	22.0	69.1	121.9	80.0	9.80	0.10	0.10	207.345
9	0.7	21.9	69.2	122.4	80.0	9.80	0.06	0.07	260.752
10	0.7	22.0	69.9	122.8	80.0	9.80	0.04	0.05	314.159
11	0.7	21.9	76.1	127.9	80.0	9.85	0.01	0.01	**********

N RUN	W*	P*	T*	P*/T*	W*T**.44	WP LBM/SEC	WS LBM/SEC	UP FT/SEC	UM FT/SEC	UU FT/SEC	UPT MACH
1	0.0	0.2667	0.9437	0.2826	0.0	3.754	0.0	213.07	71.29	71.29	0.061
2	0.1526	0.1769	0.9376	0.1886	0.1483	3.733	0.570	212.74	81.31	71.18	0.061
3	0.2593	0.1255	0.9328	0.1345	0.2515	3.745	0.971	214.28	88.98	71.69	0.061
4	0.3845	0.0706	0.9305	0.0759	0.3725	3.734	1.436	213.87	97.11	71.55	0.061
5	0.4507	0.0374	0.9294	0.0423	0.4364	3.725	1.679	213.48	101.31	71.42	0.060
6	0.4627	0.0255	0.9289	0.0274	0.4479	3.732	1.727	213.92	102.31	71.57	0.061
7	0.5285	0.0133	0.9283	0.0144	0.5115	3.714	1.963	212.97	106.20	71.25	0.060
8	0.5978	0.0093	0.9280	0.0100	0.5785	3.731	2.230	213.98	111.29	71.59	0.061
9	0.6075	0.0059	0.9272	0.0064	0.5876	3.722	2.261	213.64	111.73	71.48	0.060
10	0.6409	0.0044	0.9265	0.0047	0.6197	3.729	2.390	214.14	114.18	71.64	0.061
11	********	0.0010	0.9185	0.0011	********	3.698	*******	214.23	******	71.68	0.060

TABLE IIa PERFORMANCE DATA OF MIXING STACK (BASE DESIGN, RUN NO. 2)

DATA TAKEN ON 29 OCTOBER 1980 BY R. SHAW
D/S = 0.5
PAGE 2 OF 2

MIXING STACK PRESSURE DISTRIBUTION FOR RUN: 11 POSITION T

X/S:	0.25	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75
PMS(IN. H2O):	-0.675	-0.645	-0.240	0.390	1.125	1.460	1.295	0.610	-2.395	-0.355	0.010
PMS*:	-0.066	-0.063	-0.023	0.038	0.110	0.143	0.127	0.060	-0.234	-0.035	0.001

PLUG PRESSURE DISTRIBUTION FOR RUN: 11 POSITION T

X'/P:	0.0	0.167	0.330	0.500	0.670	0.830	1.000
PLUG (IN. H2O):	3.530	1.790	-0.720	-8.050	-0.550	0.045	1.380
PP*:	0.345	0.175	-0.070	-0.786	-0.054	0.004	0.135

MIXING STACK PRESSURE DISTRIBUTION FOR RUN: 11 POSITION S

X/S:	0.25	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75
PMS(IN. H2O):	-0.260	-0.805	-0.820	-0.615	-0.410	-0.040	0.310	0.310	0.565	0.260	0.040
PMS*:	-0.025	-0.079	-0.085	-0.060	-0.040	-0.004	0.030	0.030	0.055	0.025	0.004

PLUG PRESSURE DISTRIBUTION FOR RUN: 11 POSITION S

X'/P:	0.0	0.167	0.330	0.500	0.670	0.830	1.000
PLUG (IN. H2O):	3.530	1.360	-0.895	-5.450	-0.765	0.030	1.380
PP*:	0.345	0.133	-0.087	-0.532	-0.075	0.003	0.135

TABLE IIb AXIAL PRESSURE DISTRIBUTION DATA (BASE DESIGN, RUN NO. 2)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG)
PAGE 1 OF 2

NUMBER OF PRIMARY NOZZLES: 4
PRIMARY NOZZLE AREA (ANP): 9.05 SQ. INCHES
MIXING STACK LENGH (L): 25.50 INCHES
MIXING STACK DIMENSION (S): 8.50 INCHES
MIXING STACK L/S: 3.00

UPTAKE AREA (AU): 108.375 SQ. INCHES
AREA RATIO, AM/AP: 2.99
ORIFICE DIAMETER: 6.908 INCHES
ORIFICE BETA: 0.497
AMBIENT PRESSURE: 30.08 INCHES HG

N	POR	DPOR	TOR	TUPT	TAMB	PU-PA	PA-PS	PA-PT	SECONDARY AREA	TERTIARY AREA
RUN	INCHES OF WATER		DEGREES FAHRENHEIT			INCHES OF WATER			SQUARE INCHES	SQUARE INCHES
1	0.7	21.8	48.3	95.8	56.0	7.25	2.66	2.29	0.0	0.0
2	0.7	21.8	49.4	100.7	56.0	7.50	2.34	1.17	0.0	3.142
3	0.7	21.8	47.8	99.2	56.0	7.65	2.21	0.63	0.0	6.283
4	0.7	21.8	49.1	101.1	56.0	7.80	2.08	0.20	0.0	12.566
5	0.7	21.8	49.1	101.5	56.0	7.80	2.04	0.07	0.0	25.133
6	0.7	22.0	49.8	102.1	56.0	7.80	2.08	0.02	0.0	50.265
7	0.7	22.1	49.2	102.2	56.0	8.00	2.08	0.00	0.0	**********

SECONDARY BOX

N	W*	P*	T*	P*/T*	W*T**.44	WP	WS	UP	UM	UU	UPT MACH
RUN						LBM/SEC	LBM/SEC	FT/SEC	FT/SEC	FT/SEC	
1	0.0	0.2603	0.9283	0.2804	0.0	3.777	0.0	210.28	70.35	70.35	0.061
2	0.0	0.2254	0.9202	0.2449	0.0	3.773	0.0	211.74	70.84	70.84	0.061
3	0.0	0.2139	0.9227	0.2318	0.0	3.779	0.0	211.44	70.74	70.74	0.061
4	0.0	0.2006	0.9196	0.2182	0.0	3.774	0.0	211.82	70.87	70.87	0.061
5	0.0	0.1960	0.9189	0.2133	0.0	3.774	0.0	211.95	70.91	70.91	0.061
6	0.0	0.1983	0.9179	0.2161	0.0	3.789	0.0	213.03	71.27	71.27	0.061
7	0.0	0.1967	0.9178	0.2143	0.0	3.800	0.0	213.67	71.49	71.49	0.062

TABLE IIIa PERFORMANCE DATA OF MIXING STACK WITH MODIFIED PLUG
(RUN NO. 1, ALL PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG)
PAGE 2 OF 2

TERTIARY BOX

N	WT*	PT*	TT*	PT*/TT*	WT*TT**.44	WM	WT	UM	UE
RUN						LBM/SEC	LBM/SEC	FT/SEC	FT/SEC
1	0.0	0.2241	0.9293	0.2414	0.0	3.777	0.0	70.35	70.35
2	0.0312	0.1129	0.9202	0.1227	0.030	3.773	0.12	70.84	72.87
3	0.0455	0.0605	0.9227	0.0656	0.044	3.779	0.17	70.74	73.70
4	0.0516	0.0193	0.9196	0.0210	0.050	3.774	0.19	70.87	74.21
5	0.0610	0.0067	0.9189	0.0073	0.059	3.774	0.23	70.91	74.87
6	0.0650	0.0019	0.9179	0.0021	0.063	3.789	0.25	71.27	75.50
7	********	0.0005	0.9178	0.0005	*******	3.800	******	71.49	******

TABLE IIIb PERFORMANCE DATA OF MIXING STACK COOLING AIR FLOW (RUN NO. 1, ALL PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG)
PAGE 1 OF 2

NUMBER OF PRIMARY NOZZLES: 4
PRIMARY NOZZLE AREA (ANP): 9.05 SQ. INCHES
MIXING STACK LENGH (L): 25.50 INCHES
MIXING STACK DIMENSION (S): 8.50 INCHES
MIXING STACK L/S: 3.00

UPTAKE AREA (AU): 108.375 SQ. INCHES
AREA RATIO, AM/AP: 2.99
ORIFICE DIAMETER: 6.908 INCHES
ORIFICE BETA: 0.497
AMBIENT PRESSURE: 30.11 INCHES HG

N	POR	DPOR	TOR	TUPT	TAMB	PU-PA	PA-PS	PA-PT	SECONDARY AREA	TERTIARY AREA
RUN	INCHES OF WATER		DEGREES FAHRENHEIT			INCHES OF WATER			SQUARE INCHES	SQUARE INCHES
1	0.7	22.0	48.7	101.9	60.5	9.25	0.64	2.44	37.699	0.0
2	0.7	22.0	48.7	101.7	60.5	9.45	0.49	0.98	37.699	3.142
3	0.7	21.9	49.0	101.8	60.5	9.45	0.43	0.52	37.699	6.283
4	0.7	21.9	49.1	102.0	60.5	9.50	0.40	0.19	37.699	12.566
5	0.7	21.8	49.3	102.2	60.5	9.45	0.37	0.06	37.699	25.133
6	0.7	21.9	49.2	102.2	60.5	9.50	0.38	0.03	37.699	50.265
7	0.7	21.8	49.1	102.1	60.5	9.50	0.37	0.01	37.699	**********

SECONDARY BOX

N	W*	P*	T*	P*/T*	W*T**.44	WP	WS	UP	UM	UU	UPT MACH
RUN						LBM/SEC	LBM/SEC	FT/SEC	FT/SEC	FT/SEC	
1	0.2741	0.0619	0.9253	0.0668	0.2651	3.795	1.040	212.32	89.04	71.03	0.061
2	0.2399	0.0475	0.9256	0.0512	0.2320	3.795	0.910	212.17	86.74	70.98	0.061
3	0.2240	0.0414	0.9254	0.0447	0.2166	3.785	0.848	211.63	85.48	70.80	0.061
4	0.2173	0.0389	0.9261	0.0420	0.2101	3.785	0.822	211.67	85.05	70.82	0.061
5	0.2095	0.0362	0.9258	0.0391	0.2025	3.775	0.791	211.20	84.35	70.66	0.061
6	0.2104	0.0365	0.9258	0.0394	0.2034	3.784	0.796	211.71	84.62	70.83	0.061
7	0.2081	0.0357	0.9259	0.0385	0.2011	3.776	0.786	211.20	84.26	70.66	0.061

TABLE IVa PERFORMANCE DATA OF MIXING STACK WITH MODIFIED PLUG (RUN NO. 2, ALL PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG)
PAGE 2 OF 2

TERTIARY BOX

N	WT*	PT*	TT*	PT*/TT*	WT*TT**.44	WM	WT	UM	UE
RUN						LBM/SEC	LBM/SEC	FT/SEC	FT/SEC
1	0.0	0.2356	0.9253	0.2543	0.0	4.835	0.0	89.04	89.04
2	0.0283	0.0949	0.9266	0.1025	0.027	4.705	0.11	86.74	88.60
3	0.04[illegible]	0.0506	0.9254	0.0547	0.040	4.633	0.16	85.48	88.19
4	0.0493	0.0180	0.9251	0.0194	0.048	4.607	0.19	85.05	88.28
5	0.0562	0.0059	0.9258	0.0063	0.054	4.566	0.21	84.35	88.03
6	0.0794	0.0029	0.9258	0.0032	0.077	4.581	0.30	84.62	89.82
7	********	0.0010	0.9259	0.0011	*******	4.562	******	84.26	******

TABLE IVb PERFORMANCE DATA OF MIXING STACK COOLING AIR FLOW (RUN NO. 2, ALL PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG)
PAGE 1 OF 3

NUMBER OF PRIMARY NOZZLES: 4	UPTAKE AREA (AU): 108.375 SQ. INCHES
PRIMARY NOZZLE AREA (ANP): 9.05 SQ. INCHES	AREA RATIO, AM/AP: 2.99
MIXING STACK LENGH (L): 25.50 INCHES	ORIFICE DIAMETER: 6.908 INCHES
MIXING STACK DIMENSION (S): 8.50 INCHES	ORIFICE BETA: 0.497
MIXING STACK L/S: 3.00	AMBIENT PRESSURE: 30.11 INCHES HG

N	POR	DPOR	TOR	TUPT	TAMB	PU-PA	PA-PS	PA-PT	SECONDARY AREA	TERTIARY AREA
RUN	INCHES OF WATER		DEGREES FAHRENHEIT			INCHES OF WATER			SQUARE INCHES	SQUARE INCHES
1	0.7	21.8	49.0	102.2	61.0	9.85	0.00	2.45	**********	0.0
2	0.7	21.9	48.9	102.2	61.0	9.85	0.00	0.95	**********	3.142
3	0.7	21.8	49.0	102.4	61.0	9.85	0.00	0.17	**********	12.566
4	0.7	21.8	49.0	102.4	61.0	9.85	0.00	0.17	**********	12.566
5	0.7	21.8	49.0	102.9	61.0	9.85	0.00	0.04	**********	25.133
6	0.7	21.8	48.7	102.1	61.0	9.85	0.00	0.02	**********	50.265
7	0.7	21.8	49.0	102.2	61.0	9.85	0.00	0.00	**********	**********

SECONDARY BOX

N	W*	P*	T*	P*/T*	W*T**.44	WP	WS	UP	UM	UU	UPT MACH
RUN						LBM/SEC	LBM/SEC	FT/SEC	FT/SEC	FT/SEC	
1	********	0.0005	0.9267	0.0005	********	3.776	*******	211.08	******	70.62	0.061
2	********	0.0005	0.9257	0.0005	********	3.786	*******	211.58	******	70.79	0.061
3	********	0.0005	0.9263	0.0005	********	3.777	*******	211.15	******	70.64	0.061
4	********	0.0005	0.9263	0.0005	********	3.777	*******	211.15	******	70.64	0.061
5	********	0.0005	0.9255	0.0005	********	3.777	*******	211.34	******	70.71	0.061
6	********	0.0005	0.9268	0.0005	********	3.778	*******	211.10	******	70.63	0.061
7	********	0.0005	0.9267	0.0005	********	3.777	*******	211.08	******	70.62	0.061

TABLE Va PERFORMANCE DATA OF MIXING STACK WITH MODIFIED PLUG (RUN NO. 3, ALL PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG)
PAGE 2 OF 3

TERTIARY BOX

N RUN	WT*	PT*	TT*	PT*/TT*	WT*TT**.44	WM LBM/SEC	WT LBM/SEC	UM FT/SEC	UE FT/SEC
1	0.0	0.2400	0.9267	0.2590	0.0	*******	0.0	******	******
2	0.0278	0.0921	0.9257	0.0994	0.027	*******	0.11	******	******
3	0.0466	0.0162	0.9253	0.0174	0.045	*******	0.18	******	******
4	0.0466	0.0162	0.9253	0.0174	0.045	*******	0.18	******	******
5	0.0487	0.0044	0.9255	0.0048	0.047	*******	0.18	******	******
6	0.0562	0.0015	0.9258	0.0016	0.054	*******	0.21	******	******
7	********	0.0005	0.9267	0.0005	*******	*******	******	******	******

TABLE Vb PERFORMANCE DATA OF MIXING STACK COOLING AIR FLOW (RUN NO. 3, ALL PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG)
PAGE 3 OF 3

MIXING STACK PRESSURE DISTRIBUTION FOR RUN: 7 POSITION T

X/S:	0.25	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75
PMS(IN. H2O):	0.215	0.360	0.803	1.430	2.030	2.475	2.095	0.380	-3.080	-0.275	0.050
PMS*:	0.021	0.035	0.078	0.140	0.199	0.242	0.205	0.037	-0.302	-0.027	0.005

MIXING STACK PRESSURE DISTRIBUTION FOR RUN: 7 POSITION S

X/S:	0.25	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75
PMS(IN. H2O):	-0.100	0.165	0.415	0.780	0.795	0.790	-0.035	-0.070	0.115	0.305	0.155
PMS*:	-0.010	0.016	0.041	0.076	0.078	0.077	-0.003	-0.007	0.011	0.030	0.015

TABLE Vc AXIAL PRESSURE DISTRIBUTION OF MIXING STACK WITH MODIFIED PLUG (RUN NO. 3, ALL PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG WITH 2 ROWS OF PORTS OPEN)
PAGE 1 OF 2

NUMBER OF PRIMARY NOZZLES: 4
PRIMARY NOZZLE AREA (ANP): 9.05 SQ. INCHES
MIXING STACK LENGH (L): 25.50 INCHES
MIXING STACK DIMENSION (S): 8.50 INCHES
MIXING STACK L/S: 3.00

UPTAKE AREA (AU): 108.375 SQ. INCHES
AREA RATIO, AM/AP: 2.99
ORIFICE DIAMETER: 6.908 INCHES
ORIFICE BETA: 0.497
AMBIENT PRESSURE: 30.11 INCHES HG

N	POR	DPOR	TOR	TUPT	TAMB	PU-PA	PA-PS	PA-PT	SECONDARY AREA	TERTIARY AREA
RUN	INCHES OF WATER		DEGREES FAHRENHEIT			INCHES OF WATER			SQUARE INCHES	SQUARE INCHES
1	0.7	22.0	48.1	101.1	59.0	7.75	2.80	1.92	0.0	0.0
2	0.7	22.0	47.6	100.8	59.0	7.65	2.46	0.77	0.0	3.142
3	0.7	22.0	47.9	101.0	59.0	7.75	2.34	0.37	0.0	5.283
4	0.7	22.0	48.2	101.2	59.0	7.75	2.29	0.11	0.0	12.566
5	0.7	22.0	48.0	101.4	59.0	7.80	2.26	0.03	0.0	25.133
6	0.7	22.0	48.0	101.3	59.0	7.80	2.27	0.02	0.0	50.265
7	0.7	22.0	48.3	101.4	59.0	7.80	2.24	0.00	0.0	**********

SECONDARY BOX

N	W*	P*	T*	P*/T*	W*T**.44	WP	WS	UP	UM	UU	UPT MACH
RUN						LBM/SEC	LBM/SEC	FT/SEC	FT/SEC	FT/SEC	
1	0.0	0.2672	0.9249	0.2889	0.0	3.797	0.0	213.27	71.35	71.35	0.061
2	0.0	0.2351	0.9254	0.2541	0.0	3.799	0.0	213.08	71.29	71.29	0.061
3	0.0	0.2242	0.9251	0.2424	0.0	3.798	0.0	213.03	71.27	71.27	0.061
4	0.0	0.2194	0.9248	0.2373	0.0	3.797	0.0	213.02	71.27	71.27	0.061
5	0.0	0.2164	0.9244	0.2340	0.0	3.797	0.0	213.12	71.30	71.30	0.061
6	0.0	0.2174	0.9246	0.2351	0.0	3.797	0.0	213.09	71.29	71.29	0.061
7	0.0	0.2146	0.9244	0.2321	0.0	3.796	0.0	213.05	71.28	71.28	0.061

TABLE VIa PERFORMANCE DATA OF MIXING STACK WITH MODIFIED PLUG
(RUN NO. 1, 2 ROWS OF PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG WITH 2 ROWS OF PORTS OPEN)
PAGE 2 OF 2

TERTIARY BOX

N RUN	WT*	PT*	TT*	PT*/TT*	WT*TT**.44	WM LBM/SEC	WT LBM/SEC	UM FT/SEC	UE FT/SEC
1	0.0	0.1831	0.9249	0.1979	0.0	3.797	0.0	71.35	71.35
2	0.0251	0.0737	0.9254	0.0797	0.024	3.799	0.10	71.29	72.94
3	0.0348	0.0355	0.9251	0.0383	0.034	3.798	0.13	71.27	73.55
4	0.0379	0.0105	0.9248	0.0114	0.037	3.797	0.14	71.27	73.75
5	0.0428	0.0034	0.9244	0.0036	0.041	3.797	0.16	71.30	74.11
6	0.0647	0.0019	0.9246	0.0021	0.062	3.797	0.25	71.29	75.53
7	********	0.0005	0.9244	0.0005	*******	3.796	******	71.28	******

TABLE VIb PERFORMANCE DATA OF MIXING STACK TERTIARY AIR FLOW (RUN NO. 1, 2 ROWS OF PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG WITH 2 ROWS OF PORTS OPEN)
PAGE 1 OF 2

NUMBER OF PRIMARY NOZZLES: 4
PRIMARY NOZZLE AREA (AMP): 9.05 SQ. INCHES
MIXING STACK LENGH (L): 25.50 INCHES
MIXING STACK DIMENSION (S): 8.50 INCHES
MIXING STACK L/S: 3.00

UPTAKE AREA (AU): 108.375 SQ. INCHES
AREA RATIO, AM/AP: 2.99
ORIFICE DIAMETER: 6.908 INCHES
ORIFICE BETA: 0.497
AMBIENT PRESSURE: 30.11 INCHES HG

N	POR	DPOR	TOR	TUPT	TAMB	PU-PA	PA-PS	PA-PT	SECONDARY AREA	TERTIARY AREA
RUN	INCHES OF WATER		DEGREES FAHRENHEIT			INCHES OF WATER			SQUARE INCHES	SQUARE INCHES
1	0.7	21.9	48.6	101.6	62.0	9.15	0.69	1.99	37.699	0.0
2	0.7	21.9	47.6	101.6	62.0	9.30	0.56	0.72	37.699	3.142
3	0.7	21.8	48.7	102.3	62.0	9.30	0.50	0.39	37.699	6.283
4	0.7	21.8	48.5	101.9	62.0	9.40	0.47	0.12	37.699	12.566
5	0.7	21.8	48.5	101.9	62.0	9.35	0.46	0.03	37.699	25.133
6	0.7	21.8	48.2	101.7	62.0	9.35	0.47	0.02	37.699	50.265
7	0.7	21.8	47.7	101.5	62.0	9.35	0.45	0.00	37.699	**********

SECONDARY BOX

N	W*	P*	T*	P*/T*	W*T**.44	WP	WS	UP	UM	UU	UPT MACH
RUN						LBM/SEC	LBM/SEC	FT/SEC	FT/SEC	FT/SEC	
1	0.2838	0.0668	0.9294	0.0719	0.2748	3.787	1.075	211.77	89.51	70.85	0.061
2	0.2552	0.0541	0.9294	0.0582	0.2471	3.790	0.967	211.91	87.69	70.90	0.061
3	0.2431	0.0489	0.9283	0.0527	0.2352	3.778	0.918	211.43	86.68	70.74	0.061
4	0.2356	0.0460	0.9289	0.0496	0.2281	3.778	0.890	211.31	86.15	70.70	0.061
5	0.2318	0.0446	0.9289	0.0480	0.2244	3.778	0.876	211.30	85.90	70.69	0.061
6	0.2343	0.0456	0.9293	0.0490	0.2268	3.779	0.885	211.29	86.06	70.69	0.061
7	0.2304	0.0441	0.9296	0.0474	0.2231	3.781	0.871	211.32	85.82	70.70	0.061

TABLE VII a PERFORMANCE DATA OF MIXING STACK WITH MODIFIED PLUG (RUN NO. 2, 2 ROWS OF PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG WITH 2 ROWS OF PORTS OPEN)
PAGE 2 OF 2

TERTIARY BOX

N RUN	WT*	PT*	TT*	PT*/TT*	WT*TT**.44	WM LBM/SEC	WT LBM/SEC	UM FT/SEC	UE FT/SEC
1	0.0	0.1941	0.9294	0.2088	0.0	4.861	0.0	89.51	89.51
2	0.0242	0.0701	0.9294	0.0754	0.023	4.758	0.09	87.69	89.29
3	0.0358	0.0382	0.9283	0.0411	0.035	4.696	0.14	86.68	89.03
4	0.0397	0.0118	0.9289	0.0127	0.038	4.669	0.15	86.15	89.76
5	0.0397	0.0029	0.9289	0.0032	0.036	4.654	0.15	85.90	88.50
6	0.0648	0.0020	0.9293	0.0021	0.063	4.665	0.24	86.06	90.32
7	********	0.0005	0.9296	0.0005	*******	4.652	******	85.82	******

TABLE VIIb PERFORMANCE DATA OF MIXING STACK TERTIARY AIR FLOW (RUN NO. 2, 2 ROWS OF PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG WITH 2 ROWS OF PORTS OPEN)
PAGE 1 OF 3

NUMBER OF PRIMARY NOZZLES: 4
PRIMARY NOZZLE AREA (AMP): 9.05 SQ. INCHES
MIXING STACK LENGH (L): 25.50 INCHES
MIXING STACK DIMENSION (S): 8.50 INCHES
MIXING STACK L/S: 3.00

UPTAKE AREA (AU): 108.375 SQ. INCHES
AREA RATIO, AM/AP: 2.99
ORIFICE DIAMETER: 6.908 INCHES
ORIFICE BETA: 0.497
AMBIENT PRESSURE: 30.12 INCHES HG

N	PUR	OPOR	TOR	TUPT	TAMB	PU-PA	PA-PS	PA-PT	SECONDARY AREA	TERTIARY AREA
RUN	INCHES OF WATER		DEGREES FAHRENHEIT			INCHES OF WATER			SQUARE INCHES	SQUARE INCHES
1	0.7	21.8	47.8	101.4	63.0	9.80	0.00	1.91	**********	0.0
2	0.7	22.0	48.1	101.4	63.0	9.90	0.00	0.66	**********	3.142
3	0.7	22.0	47.6	101.4	63.0	9.95	0.04	0.34	**********	6.283
4	0.7	22.0	47.7	101.3	63.0	9.90	0.01	0.09	**********	12.566
5	0.7	22.0	47.8	101.4	63.0	9.95	0.00	0.03	**********	25.133
6	0.7	22.0	48.0	101.5	63.0	9.90	0.00	0.01	**********	50.265
7	0.7	22.0	48.2	101.5	63.0	9.90	0.00	0.00	**********	**********

SECONDARY BOX

N	W*	P*	T*	P*/T*	W*T**.44	WP	WS	UP	UM	UU	JPT MAC-1
RUN						LBM/SEC	LBM/SEC	FT/SEC	FT/SEC	FT/SEC	
1	********	0.0005	0.9316	0.0005	********	3.782	*******	210.99	******	70.59	0.061
2	********	0.0005	0.9316	0.0005	********	3.798	*******	211.89	******	70.89	0.061
3	********	0.0039	0.9316	0.0042	********	3.800	*******	212.01	******	70.93	0.061
4	********	0.0010	0.9317	0.0010	********	3.799	*******	211.94	******	70.91	0.061
5	********	0.0005	0.9316	0.0005	********	3.799	*******	211.96	******	70.91	0.061
6	********	0.0005	0.9314	0.0005	********	3.798	*******	211.95	******	70.91	0.061
7	********	0.0005	0.9314	0.0005	********	3.797	*******	211.91	******	70.90	0.061

TABLE VIIIa. PERFORMANCE DATA OF MIXING STACK WITH MODIFIED PLUG (RUN NO. 3, 2 ROWS OF PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG WITH 2 ROWS OF PORTS OPEN)
PAGE 2 OF 3

TERTIARY BOX

N RUN	WT*	PT*	TT*	PT*/TT*	WT*TT**.44	WM LBM/SEC	WF LBM/SEC	UM FT/SEC	UE FT/SEC
1	0.0	0.1879	0.9316	0.2017	0.0	*******	0.0	******	******
2	0.0231	0.0644	0.9316	0.0691	0.022	*******	0.09	******	******
3	0.0329	0.0326	0.9316	0.0350	0.032	*******	0.13	******	******
4	0.0342	0.0088	0.9317	0.0094	0.033	*******	0.13	******	******
5	0.0394	0.0029	0.9316	0.0031	0.038	*******	0.15	******	******
6	0.0455	0.0010	0.9314	0.0010	0.044	*******	0.17	******	******
7	********	0.0005	0.9314	0.0005	*******	*******	******	******	******

TABLE VIIIb PERFORMANCE DATA OF MIXING STACK COOLING AIR FLOW
(RUN NO. 3, 2 ROWS OF PORTS OPEN)

DATA TAKEN ON 1 DECEMBER 1980 BY R. SHAW
L/S = 3.0 (DATA ON MODIFIED PLUG WITH 2 ROWS OF PORTS OPEN)
PAGE 3 OF 3

MIXING STACK PRESSURE DISTRIBUTION FOR RUN: 7 POSITION T

X/S:	0.25	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75
PMS(IN. H2C):	0.040	0.125	0.530	1.155	1.795	2.130	1.750	0.495	-2.830	-0.240	0.070
PMS*:	0.004	0.012	0.052	0.113	0.175	0.208	0.171	0.048	-0.276	-0.023	0.007

MIXING STACK PRESSURE DISTRIBUTION FOR RUN: 7 POSITION S

X/S:	0.25	0.50	0.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75
PMS(IN. H2C):	-0.115	-0.020	0.145	0.440	0.480	0.625	0.345	0.120	0.265	0.390	0.235
PMS*:	-0.011	-0.002	0.014	0.043	0.047	0.061	0.034	0.012	0.026	0.038	0.023

TABLE VIIIc AXIAL PRESSURE DISTRIBUTION OF MIXING STACK WITH MODIFIED PLUG (RUN NO. 3, 2 ROWS OF PORTS OPEN)

APPENDIX A

FORMULAE

Presented here are the formulae used to obtain the primary and secondary mass flow rates. According to the ASME Power Test Code [5], the general equation for mass flow rate appearing in equation (a)

$$W(\text{lbm/sec}) = (0.12705)\ K\ A\ Y\ F_a\ (\rho\ \Delta P)^{0.5} \qquad \text{(a)}$$

may be used with flow nozzles and square edge orifices provided the flow is subsonic. In the above equation, K (dimensionless) represents the flow coefficient for the metering device and is defined as $K = C(1-\beta^4)^{-0.5}$ where C is the coefficient of discharge and β is the ratio of throat to inlet diameters; $A(in^2)$ is the total cross sectional area of the metering device; Y (dimensionless) is the expansion factor for the flow; F_a (dimensionless) is the area thermal expansion factor; $\rho(lbm/ft^3)$ is the flow mass density; and ΔP (inches H_2O) is the differential pressure across the metering device. Each of these quantities are evaluated, according to the guidelines set forth in Ref. 5, for the specific type of flow measuring device used.

Using a square edge orifice for measurement of the primary mass flow rate, the quantities in equation (a) are defined as follows:

1. The flow coefficient K is 0.62 based on a β of

0.502 and a constant coefficient of discharge over the range of flows considered of 0.60.

2. The orifice area is 37.4145 in^2.
3. Corresponding to the range of pressure ratios encountered across the orifice, the expansion factor Y is 0.98.
4. Since the temperature of the metered air is nearly ambient temperature, thermal expansion factor is essentially 1.0.
5. The primary air mass density ρ_{or} is calculated using the perfect gas relationship with pressure and temperature evaluated upstream of the orifice.

Substituting these values into equation (a) yields

$$W_p(\text{lbm/sec}) = (2.882) \quad (\rho_{or}\,\Delta P_{or})^{0.5} \qquad \text{(b)}$$

The secondary mass flow rate is measured using long radius flow nozzles for which case the quantities in equation (a) become:

1. For a flow nozzle installed in a plenum, β is approximately zero in which case the flow coefficient is approximately equal to the coefficient of discharge. For the range of secondary flows encountered, the flow coefficient becomes 0.98.
2. A is the sum of the throat areas of the flow nozzles in use.
3. Since the pressure ratios across the flow nozzles

are very close to unity, the expansion coefficient Y is 1.0.

4. Since the temperature of the metered air is nearly ambient temperature, the thermal expansion factor is essentially 1.0.

5. The secondary air mass density ρ_s is evaluated using the perfect gas relationship at ambient conditions.

Substituting these values into equation (a) yields the equation for the secondary mass flow rate measured using long radius flow nozzles.

$$W_s(\text{lbm/sec}) = (0.13451)\ A\ (\rho_s \Delta P_s)^{0.5} \qquad \text{(c)}$$

APPENDIX B

CALCULATION OF THE MIXING STACK PRIMARY DIMENSION

The primary dimension of the mixing stack geometry was obtained from the mass rate of flow equations. In order to use the same mass flow conditions as used in the previous tests with cylindrical mixing stacks, the cross-sectional entrance area of the rectangular mixing stack of this study was made equal to that of the cylindrical models. By ensuring that the pressure differential across the orifice was the same as in previous testing, the uptake velocities would then be equal and the mass flow equation reduces to an equation relating the areas of the models, equation (a).

$$A_{\text{Rectangular}} = A_{\text{Circular}} \qquad \text{(a)}$$

Substitution of the mathematical relations for each area results in the general equation, equation (b)

$$ab = 0.25\,\pi\,d^2 \qquad \text{(b)}$$

With a side to side ratio of 1.5 to 1 for the rectangle, the general equation yields equation (c).

$$(1.5s)(s) = 0.25\,\pi\,d^2 \qquad \text{(c)}$$

Further reduction of equation (c) gives the relation between the primary dimension (s) and the diameter (d), equation (d).

$$s = 0.724\,d \qquad \text{(d)}$$

This relation was employed in the design of the eductor model using the diameter of Lemke and Staehli [7].

APPENDIX C

UNCERTAINTY ANALYSIS

The determination of the uncertainties in the experimentally determined pressure coefficients, pumping coefficients, and velocity profiles was made using the method described by Kline and McClintock [6]. The uncertainties obtained by Ellin [1] using the second order equation suggested by Kline and McClintock [6] are all applicable to the experimental work reported herein and are summarized in the following table.

TABLE XIV

UNCERTAINTY IN MEASURED VALUES

T_s	±1 R
T_p	±1 R
P_a	±0.01 psia
ΔP	±0.01 in. H_2O
P_V	±0.01 in. H_2O
P_u	±0.05 in. H_2O
$\Delta P_s(+)$	±0.01 in. H_2O
$\Delta P_t(++)$	±0.01 in. H_2O
P_{or}	±0.01 in. H_2O
ΔP_{or}	±0.20 in. H_2O
T_{or}	±1 R

T_a	±1 R
PT (+++)	±0.1 in. H_2O

UNCERTAINTY IN CALCULATED VALUES

$\frac{\Delta P^*}{T^*}$	1.9%
$W^*T^{*0.44}$	1.4%
V/V_{avg}	2.5%

(+) The pressure differential across the secondary flow nozzles, P_s, is the major source of uncertainty in the pumping coefficient.

(++) The pressure differential across the tertiary flow nozzles, P_t, is the major source of uncertainty in the pumping coefficient.

(+++) The measurement of the total pressure for the velocity profile is the major source of uncertainty in the velocity calculation.

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